HEATING AND VENTILATING

INCLUDING AIR CONDITIONING

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Unit Ventilators of Schools . . page 79

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9 reasons why...

1. **MORE ACTUAL CAPACITY PER DOLLAR** due to Armstrong's unique free floating lever mechanism.

2. **NO GAMBLE IN SELECTING TRAPS** because Armstrong capacity ratings are actual trap tests with hot condensate instead of slide rule calculations, cold water, or orifice tests.

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HEATING AND VENTILATING

A MONTHLY MAGAZINE FOR ENGINEERS AND CONTRACTORS CONCERNED WITH THE DESIGN AND APPLICATION OF AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION IN BUILDINGS

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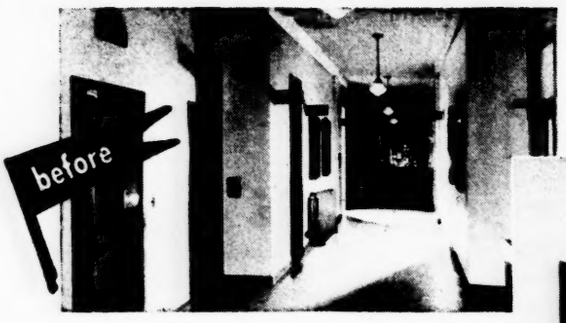
PUBLISHERS OF MACHINERY & MACHINERY'S HANDBOOK
The INDUSTRIAL PRESS

THIS MONTH'S COVER

Section of the knitting room in the new $1.5-million textile engineering building of the A. French Textile School at Georgia Tech. Building is completely air conditioned with Niagara units. Air conditioning was designed by C. T. Baker, Atlanta consulting engineer. Photo by Joseph W. Molitor, Anemostat Corp. of America.

MODERN



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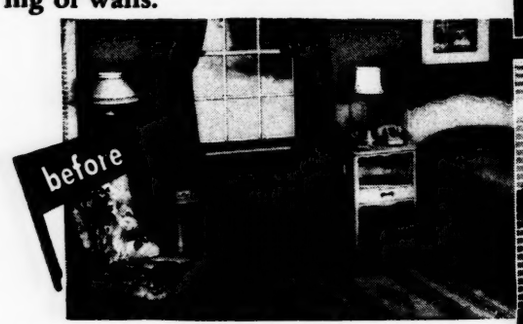
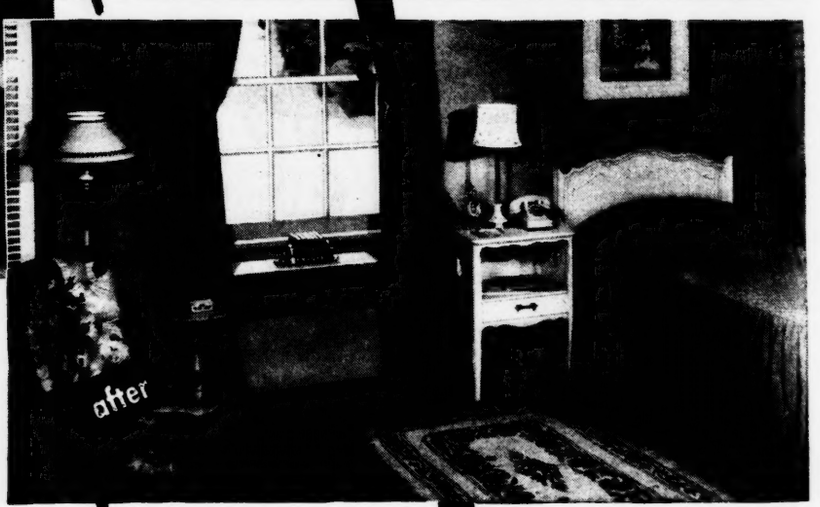
★ Puzzle . . . find the heating equipment! Instead of old, bulky, dirt-collecting radiators, new Fedders Baseboard Radiation surrounds everyone with comfort. No interference with draperies or furniture placement. Fedders exclusive anti-streak design prevents streaking of walls.





★ See how Fedders Convector-Radiators changed the complexion of this bedroom by making a smart new console unit take the place of an old-fashioned radiator.

Washington News

LORING F. OVERMAN

WHEN members of Congress recessed on September 24 for an announced two months, it might as well have announced that "October is to be question-mark month."

For one-third of the Senators and all members of the House, the question is: Will constituents send them back after the November elections?

For the rest of America the questions will be legion:

Will favorable progress in Korea continue?

If so, will hostilities end there, or break out somewhere else?

How will the new defense production controls affect me?

Will there be price and wage controls?

Will there be shortages of material? Of labor? Of orders?

How did we get into another war, anyway?

No Question-Tax

One thing that had been keeping Congress from adjourning, recessing, or otherwise going home was the President's insistence that there be a new tax bill —something to start paying for the new military program as we go along. Dutifully, Congress completed a $4.7 billion tax bill on September 23—calling for new and heavy obligations for individuals and corporations. It ducked the idea of imposing new taxes on excess profits. Instead it "directed" the next Congress to take up the question of excess profits taxes as an early order of business when it returns on November 27. This procedure was in itself somewhat unusual, and it remains to be seen whether one Congress can pledge its successor organization to a specific action.

Meanwhile, starting on October 1, business and individuals began paying for the present military involvement. Some 50 million individual taxpayers will contribute almost $3 billion more a year. Corporation normal rates are boosted from a top of 38% to 45%, to bring in another $1.5 billion a year. Some $200 million more will be gathered by plugging loopholes and adding certain excises.

In signing the measure, President Truman commented that it was only a beginning, and that he will expect the House Ways and Means committee to have another bill ready for early action when Congress returns. The second bill would raise another $4 billion to $7 billion—bringing federal taxes to an annual total of more than $50 billion—a figure much higher than for any single year during World War II.

Corporations, in addition to their rate increases, are required to speed up on the payment of their taxes, bringing an additional $5 billion into the Treasury in the next five years. The new law also imposes a new 10% manufacturers' excise tax on television sets. Also provided for are tax exemptions for servicemen fighting in Korea, and reductions for their officers. Tax incentives for industry to retool for production of materials needed by the military are included.

After Korea?

As September ended, developments in Korea were sufficiently favorable for Washington observers to predict that the worst there would soon be over. At best, however, all were anticipating long-drawn-out mopping-up actions; continued policing operations, and the inevitable rehabilitation of the fallen foe and the refugees.

Consequently, no one was forecasting any early braking of the defense production program, or any reduction in the demand for military personnel. Most optimistic guesses were that the indicated successful conclusion of the Korean campaign would deter any enemy from early offensives elsewhere, but that the need for building up defense strength would continue until the Red menace is finally neutralized—one way or another.

This latter appraisal—that preparations for a showdown must continue—is responsible for countless developments almost daily in Washington. Among recent actions were the following:

1. Congress passed a Defense Production Act giving the President powers to designate agencies to control production, ration commodities, control prices and wages.

2. An Executive Order (No. 10161) was issued by the President naming the agencies to assist the Chief Executive, and setting up an Economic Stabilization Agency, a new outfit with power to govern wages and prices.

3. W. Stuart Symington, chairman of the National Security Resources Board, was named coordinator of the entire mobilization effort.

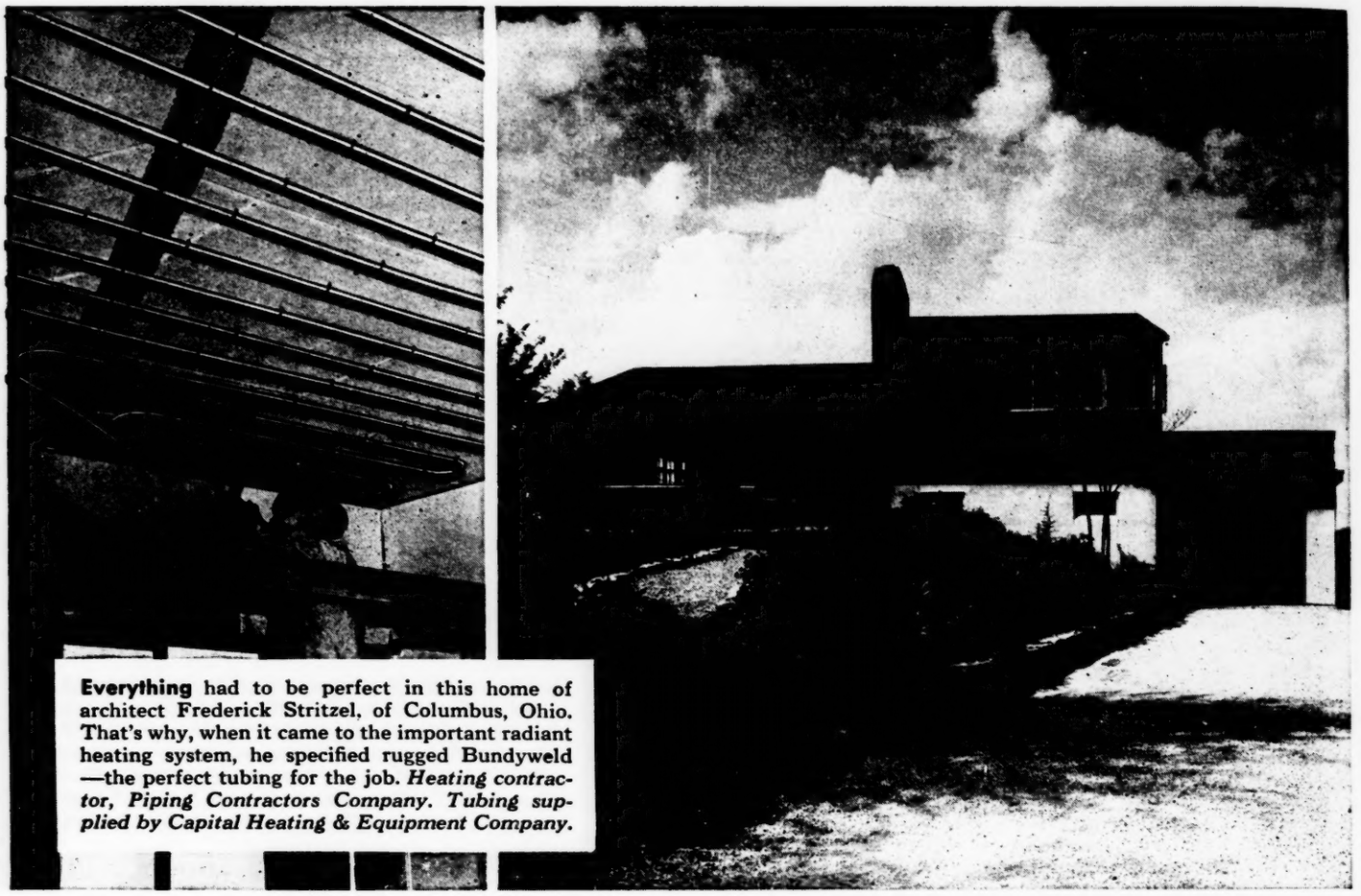
4. The President issued an Executive Order (No. 10160) requiring the preservation of business records for the period May 24, 1950 to June 24, 1950, as a guide to be taken into consideration later if it is decided to impose price controls.

5. The Federal Reserve System reinstituted regulation W, effective September 18, limiting down payments and maximum maturities for the purchase of numerous items. For automobiles the down payment is to be 33 1/3%, to mature in 21 months; appliances. 15% and 18 months; furniture and rugs, 10% and 18 months; home repairs, 10% and 30 months.

6. Commerce Secretary Sawyer, in order No. 123, established the National Production Authority and named William Henry Harrison, president of the International Telephone and Telegraph Corporation, as NPA Administrator. The Commerce Department also opened a series of industry meetings to study supply prospects.

7. Following the initial conference with steel executives, 39 iron and steel mill products were placed under export controls, effective September 30.

8. On September 18, NPA issued an inventory control order covering seven classes of important materials—building materials, chemicals, forest products,



Everything had to be perfect in this home of architect Frederick Stritzel, of Columbus, Ohio. That's why, when it came to the important radiant heating system, he specified rugged Bundyweld —the perfect tubing for the job. *Heating contractor, Piping Contractors Company. Tubing supplied by Capital Heating & Equipment Company.*

Architect picks Bundyweld for radiant heating in own home

It's top praise for top tubing when professionals like F. Stritzel pick Bundyweld Tubing for radiant heating systems in their own homes.

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iron and steel, metals and minerals, rubber materials, and textiles of specified types. Intentionally—at least until after the elections—the controls were extremely mild. The regulation limits to "a practicable minimum working inventory," certain materials and metals which industrial managers may order or have on hand.

Intent of the inventory control regulation is to require certain voluntary cooperation, and to imply that if this is not forthcoming, something more specific and more drastic can be expected. The regulation sets no hard and fast rule as to what constitutes a "practicable minimum working inventory," indicating that trade practices or production scheduling will serve as a suitable guide.

Analysts of the order are betting that a few chiselers will upset the boat in a relatively short time, and that more drastic controls will be announced shortly.

Controlling Manpower

The inventory control order—or its probable successor, rationing and priority orders—are being looked to as an acceptable means of channeling manpower into the war effort. It is observed that the American people will accept the shunting of materials out of civilian enterprises into war plants, but that they would resist any effort to insist that a worker be similarly moved around.

Therefore, if workers are needed in tank factories, for example, there is one sure way of getting a worker to quit making automobiles and start building tanks. The government merely curtails the flow of materials to the auto factory and diverts it to the tank factory. The worker follows the material and the job. So it may be expected that if available supplies are not shared willingly, or if the war program requires more manpower or facilities, NPA will not hesitate to swing a big stick instead of its present riding crop.

Cutback Picture

In seeking to evaluate the possible effects of cutbacks, it must be remembered steel output is setting new records this year at around 71 million tons of products, as contrasted with 39 million in 1939. Also, there are indications that production will rise to 74 million tons in 1951.

Military needs in the fiscal year 1951 have been officially estimated at 4 million product tons. This covers only our own direct military needs, excluding the steel which will enter into military items for Atlantic Pact nations, and indirect needs associated with stepped-up military programs.

An unofficial trade estimate is that over-all government requirements by July, 1951, might reach a rate of as much as 12 million tons of ingots. Adjusted to a product basis, this would be about 9 million tons. Military uses averaged 34.2 million tons per year during the three full years of War II (1942-44). About 50% of this, however, was for ships and war construction, neither of which will require much steel under military programs now scheduled.

Civilian use of steel is setting a new record, approximately 65 million tons this year. If next year's capacity rises by 3 million tons, if exports remain the same, and if the military requires 9 million tons, some 61 million would remain available for civilian use. With the exception of this year's record-breaker, a 61-million ton total would be larger than any previous annual civilian use.

Because of the shapes required by the military, the balance available to civilians might not be evenly distributed over all types of products. Civilian industries in most favorable positions would include railway equipment, farm and other machinery and tools, and oil field supplies.

Oil and gas transmission and oil processing will be on the list of preferred customers. More pipe lines would ease the defense load on the railroads in case of an all-out emergency, and would help prevent fuel oil shortages. It is not considered likely that the over-all use of steel in the oil, gas, water, and mining industries will equal this year's all-time high of 6.1 million tons.

Saving of steel for the military is expected to be most noticeable in the auto and construction industries. Together they are taking an estimated 26.2 million tons, or about 37% of the total. If these industries were cut to 1949 levels, 5.7 million tons would be released—approximately 80% of the maximum estimated increase in military needs. It should also be remembered that heaviest war needs will be for light flat-rolled products—the kind used in automobiles, refrigerators and home appliances.

The question of who gets how much steel is indeed a tough one, at this stage of the game. Government asks industry how much it can cut consumption and increase production; industry inquires how much is to be needed for what military purposes, and everyone faces the Kremlin and wonders.

All-Out Programs

While this question remains unanswered, indications are that preparations are being made in Washington to do too much, rather than too little. Secretary of Defense Johnson has been sacrificed on the altar of mistaken military planning, and those now assuming the reins seem to sense that America would rather buy too many guns than have trainees carry wooden substitutes.

Passage of the anti-Communist bill over the President's veto is a typical example. Congress has sensed that the folks back home are sufficiently aroused to demand some sort of legislation to curb the spread of Communism in this country. Although the present legislation may not do all that is expected of it, members of Congress hesitated to face their constituents without having passed some sort of a bill intended to do a job on the Reds.

Thus, the families whose sons are being drafted are more interested in seeing his outfit well-equipped, than in worrying about a new car for themselves or for a neighbor. So plans in Washington are being made with an eye to the military possibility, and with not too much consideration for the civilian reaction. Full force of this reasoning may not show up until after the votes have been counted, but there is a lot of midnight oil being burned in an effort to see that Secretary Marshall's new Department of Defense is well equipped and well manned.

BRIEFLY STATED

• Dr. Clarence A. Mills, professor of experimental medicine at the University of Cincinnati, recently announced the formal opening of his experimental house, "Reflection Point," located at 2311 Fairview Avenue in Cincinnati. From September 20 to 29, the house was open for inspection by special arrangement only. According to Dr. Mills, the house has now operated satisfactorily at outside temperatures ranging from 20F in April to 96F in June. Unique features of the house include reflective aluminum surfaces on walls and ceilings said to permit low air temperatures for comfort with very little space heating. Both heating and cooling are radiant from sources recessed in walls near ceilings.

• Stephen L. Fitzpatrick, a Detroit inventor, has worked out a way to use powdered coal as a home fuel. In the fall, he plans to market a small furnace that will burn powdered coal suspended in oxygen, mixing it with air the way gasoline mixes in the carburetor of an automobile. Mr. Fitzpatrick has formed a company, BloKolGas System, Inc., to manufacture and distribute both furnace and fuel. He'll start out making a conversion-type unit for commercial use, and will follow up with smaller units.

• Completing more than two years of intensive design, product engineering and laboratory testing work, The Trane Co. has announced development of a completely new and expanded line of centrifugal fans. The company says that the new centrifugal fans can deliver a required volume of air with horsepower as much as 23% lower than other fans carrying the same rating.

• Dr. Harold Vagtborg, president of Southwest Research Institute, has announced that Crane Co. of Chicago has joined with Revere Copper and Brass, Inc., in co-sponsorship of the Institute's Quality House Program. During the past two years the nationwide program to assist merchant builders in providing house buyers with top-quality homes at moderate prices has approved plans and specifications for more than 1,700 houses built in projects from coast to coast.

• Building materials production reached an all-time high in May 1950, according to a report released by the U. S. Department of Commerce. Output of lumber, hardwood flooring, cement, softwood plywood, and gypsum board was substantially above earlier postwar peaks. Almost twice as much cement was produced in May this year as in the average month before the war.

• The Stratfield Hotel, in Bridgeport, Conn., this winter will have in operation a snow removal system in its main driveway, at the hotel's automobile entrance and service entrance and in concrete sidewalk. It is one of the largest snow melting systems in the eastern seaboard states using wrought iron pipe coils. It was designed by Westcott and Mapes, Inc., New Haven architects and engineers.

• President George D. Stoddard of the University of Illinois recently granted an additional year's leave of absence to William H. Scheick, executive director of BRAB, in order that he might continue to pilot BRAB activities. Mr. Scheick is director of the Small Homes Council of the University of Illinois. While he is on active duty with BRAB, James T. Lendrum is acting director of the Small Homes Council.

• Texas Gas Transmission Corp. reported that operating revenues during the first six months of 1950 amounted to $13,677,000 as compared with $5,281,000 in the corresponding period of 1949. For the six months ended June 30, the company's net income totalled $1,877,000 or 84.3 cents a share on outstanding common stock. This compares with a net income of $1,027,000 or 46.2 cents per share in the same period a year ago.

• A new air pollution ordinance in Providence, R. I., states that bituminous coal may not be hand-fired in any heating or power plant.

• John F. McMahon, administrative fellow of the Mellon Institute, died suddenly in his office on Friday, September 15. At the time of his death he was rounding out his fourteenth year as a member of the Institute and his eighth as managing director of the Industrial Hygiene Foundation. Born in South Waverly, Pa., in 1903, John McMahon attended schools at Sayre, Pa., and then graduated with a B.A. from the University of Notre Dame in 1928. From 1929 to 1931, he served as a reporter on the *New York Sun*. For the next two years he was assistant to James T. Grady, public relations director for Columbia University, the American Chemical Society, and other technical organizations. In 1932 he went to Pittsburgh as night editor of the Associated Press. After a four-year stint with the AP, he joined the Industrial Hygiene group at Mellon Institute to attend to public relations. Six years later he succeeded the late Dr. H. B. Meller as managing director of the Foundation.

• Abnormally warm weather was one of the most important contributing factors in the sharp decline in the 1949 production of Pennsylvania anthracite to 42,701,724 net tons, a decrease of 25% from 1948. Other important factors contributing to the decrease in output were competition from other fuels, especially fuel oil and natural gas, and loss in foreign markets. With increased output of coal in Europe, those countries were in a better position to supply their fuel needs, and Canada's requirements dropped commensurately with the extremely warm weather of the 1949 winter months.

• For the first time in eighty-three years of annual convention meetings, The American Institute of Architects will include a selected product exhibit at its 1951 convention, to be held in May of next year at the Edgewater Beach Hotel in Chicago. Only those building products or methods recently placed in production or shortly to be offered commercially will be granted space. These must be capable of achieving improved construction at moderate cost. The number of exhibits will be limited to fifty.

INFRA-RED HEAT RAYS PIERCE 331 MILES OF DUST & FOG

"Aerial photographs taken by Capt. Albert Stevens . . . With infra-red sensitized photographic plates, views of mountains as far as 331 miles distant from the photographer have been taken, though the mountain itself was invisible to the eye because of the intervening haze and fog." From booklet, "Some Reflection and Radiation Characteristics of Aluminum,"* published by American Society of Heating and Ventilating Engineers.

"A mirror, consisting of glass with a silvered surface on the back of the glass, is an excellent reflector of light but it is a very poor reflector of infra-red radiation corresponding to room temperature. In fact, such a mirror would have about the same reflectivity for infra-red as a heavy coating of black paint. With this in view, it is obviously impossible to judge the infra-red reflectivity or emissivity of a surface by its appearance to the eye." From booklet, "Thermal Test Co-efficients of Aluminum Insulation for Buildings,"* published by American Society of Heating and Ventilating Engineers.

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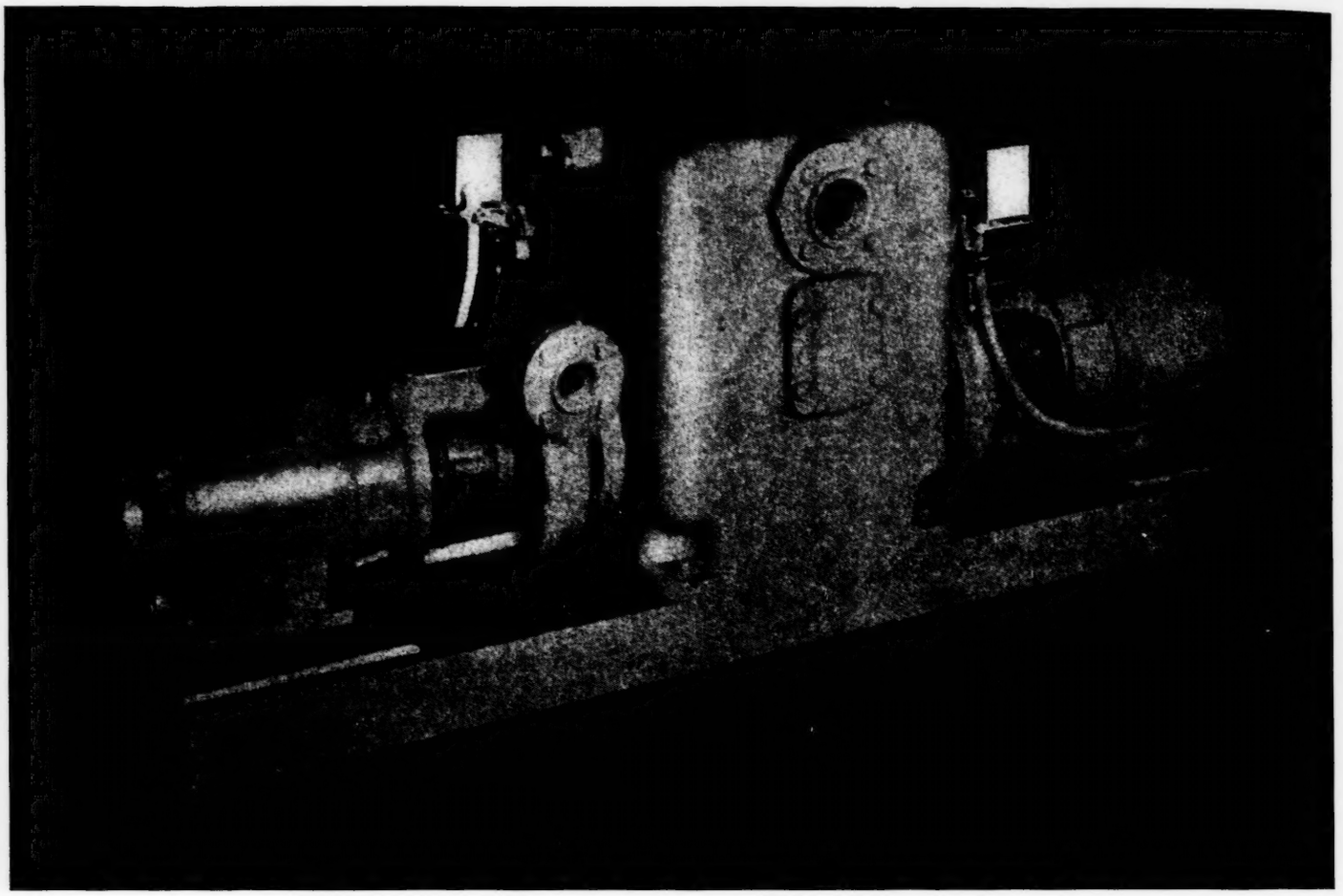

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Designed for steam systems on heating, drying, or process work, these pumps offer the most reliable and efficient means of returning hot condensate to boiler or hot well. In many cases these pumps will save costly installation of boilers in a pit to provide gravity flow of returns from basement radiators and heating devices.

The Jennings Condensation Pumps are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Pumps are bronze fitted throughout, with tobin bronze shaft. The pumping impeller is of special design, made possible by our wide experience with return line heating pumps, and is especially adapted for handling hot water with the greatest possible efficiency.

Full automatic control is furnished by means of a ball float and float switch mechanism mounted on the receiving tank. All wiring is made up at the factory, and it is necessary only to connect the pump to the system and hook up the leads. Complete information regarding these economical pumps sent promptly on request.

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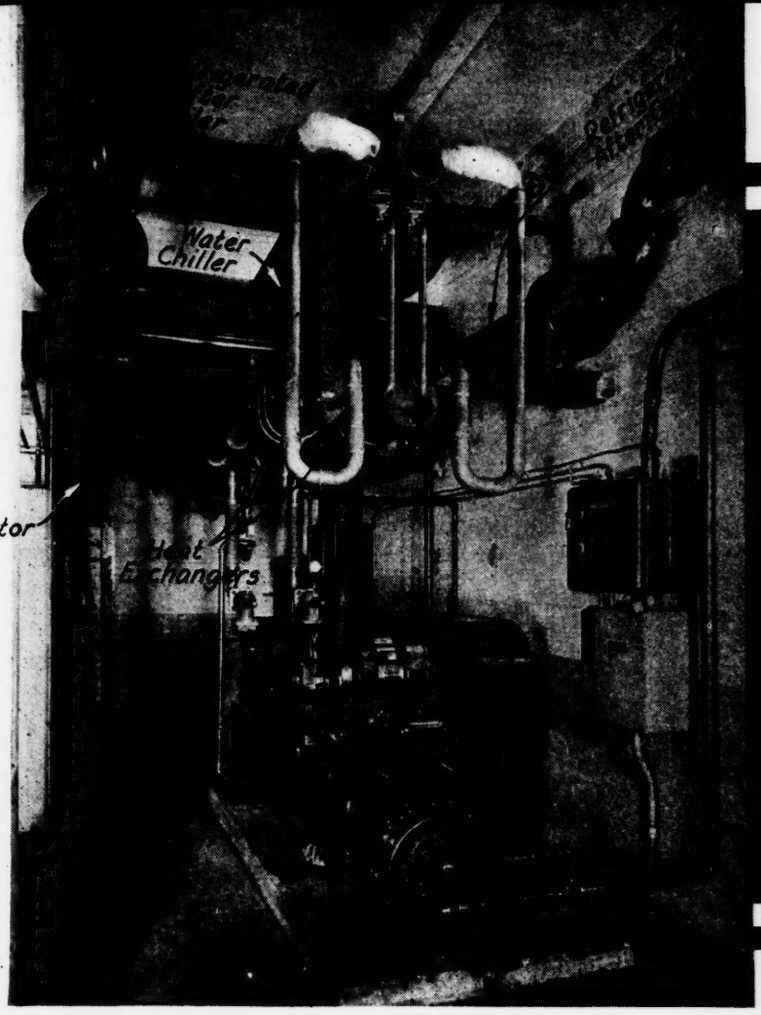

Refrigeration Removes Moisture from Compressor Air Lines

J. PARTINGTON, JR.
Supervisor Industrial Sales, Air Conditioning Dept.
General Electric Co., Bloomfield, N. J.

Fig. 1. End view of the refrigeration system.

Moisture in compressed air lines reduces the normal life of the piping and influences the operation of compressed air tools. An engineering analysis is presented to show the size of a refrigerating unit required to avoid moisture condensation in a typical design.

REFRIGERATION is important as a means of removing moisture from the compressed air supply lines in industrial plants where compressed air is used for motive power and control. Moisture, which is always present in the air delivered to compressors, collects in compressed air supply lines unless steps are taken to correct this condition. Where there is water in the air lines, there may be interruption in the operation of machine tools.

Unfortunately, compressed air has been taken so much for granted in an industrial plant that it is often looked upon as something that will always be available in the proper amount and quality. This supply is not infallible and before many shut-downs occur, some investigation should be made of the economics of refrigeration in this industrial application.

To obtain a better picture of the value of refrigeration in removing this accumulated moisture from the compressed air lines let us trace what happens to air in a typical air compressor.

Effects of Moisture

If atmospheric air at 90F dry bulb and 75F wet bulb (relative humidity, 50%) enters the compressor inlet, this entering air carries with it 106 grains of moisture per pound of dry air. If the temperature of this air is kept constant and its absolute pressure is doubled, thus reducing the volume one half, its moisture-carrying capacity is reduced in the same proportion. Thus our air of 50% relative humidity becomes 100% R.H. air, or saturated, when its absolute pressure is doubled. If its pressure be raised to 90 lb gage, (105 lb absolute pressure) the humidity becomes 350% (7 $\times$ 50%) of which 250% will be condensed out.

While we have assumed that the air temperature would remain constant, this is far from actual fact. As air is compressed, its temperature rises rapidly. Each 20F rise in temperature approximately doubles its capacity to hold moisture. Air entering a single-stage compressor at 90F and compressed to 100 lb gage, will usually have a delivery temperature well in excess of 300F.

As this 100 lb pressure air flows into the transmission line, however, its temperature drops, its moisture-carrying capacity is reduced, and the excess moisture condenses out into water in the line. If the line is long enough, the water will form a stream in the bottom of the line to be carried along with the air current. The use of an aftercooler, cooled by city water, will reduce the amount of free water in the transmission line. However, with average city water temperatures available for cooling, there is still much water left in the air, to be condensed in the line.

Now that we understand how the water gets into the line, let us see the effect of this moisture. Water in the line is carried into the tools or machines where the air is being used and it washes away the lubricating oil. Lack of lubrication causes excessive wear, which brings high maintenance expense. Deprived of adequate lubrication, sluggish operation of tools or machines results, with noticeable lowering of efficien-

cies. Such an effect is especially pronounced in high speed tools where wearing surfaces are limited in size, and where excessive wear creates air leakage and poor operational economy. At 6 cents per 1,000 cu ft of 100 lb air, a 1/16 inch opening will waste approximately $11 worth of air per month; a 1/8 inch opening, $44 per month.

Water in air lines can cause corrosion and rusting of parts being manufactured, where air is used to blow away chips or dust. The oil vapors will also prove objectionable when parts are to be hand or spray painted later on.

Due to the release of pressure at the point of use, air can freeze and interfere with the proper operation of tools or mechanisms. This freezing of water in air lines can burst pipes. It can reduce the carrying capacity of the lines and produce water hammer by accumulating moisture at low points in the line. Oil vapor from the compressor coagulates on cooling and settles out in the lines. This, when combined with water, exerts a corrosive effect on gaskets, air hose, and the pipe itself, and results in increased maintenance and operational outages.

Of prime importance is the damage done to air piping; much of this piping is underground and is hard to locate. The replacement cost of such an investment is too expensive to even contemplate.

The following example, while unusual in this country, is introduced to show the importance of the moisture problem particularly during high humidity weather, and to indicate the need for cooling immediately adjacent to the compressor and before the air enters the distribution lines.

Let us take 1,000 cu ft of free air at sea level atmospheric pressure, 80F temperature and 100% R.H. This amount of air will hold 1.58 lb of moisture. Compress this air to 100 lb per sq in. and cool it back down in an aftercooler to its entering temperature of 80F, and approximately 87% of its moisture will be condensed out. In operating a 1,000 cfm compressor under these conditions you would condense approximately 10 gph of water in your system. You could achieve this kind of operation if you had approximately 180 gpm of 70F water available to cool the air back down to a temperature of 80F.

Let us examine an actual case and see what we are concerned with in terms of refrigeration horsepower when it comes to cooling compressed air.

The air compressor has a capacity of 1,500 cfm of 100 lb per sq in. air, approximately 11,000 cfm of free air at 95F. The compressor itself has water-cooled cylinders, a water-cooled intercooler, and a water-cooled aftercooler. The aftercooler has reduced the compressed air temperature to 100F and the air is saturated. The air piping runs underground from the compressor to the various plant buildings, some of which are as much as 1,200 ft away. The piping has been properly laid to avoid freezing so that the earth temperature around it is usually 50F at all times of the year. That being true, the air in the lines will probably never be cooled much below 60F. To be sure, however, that no moisture is dropped out in the piping, we will assume a desired leaving condition of 55F and saturated from the air cooler.

In order to figure our cooling load, we must figure the number of pounds of air being handled and the moisture content of the air at the entering and leaving conditions at the air cooler. To find the pounds of air being handled we must find V first, which is the volume in cubic feet occupied by 1 lb of dry air plus water vapor. V is calculated from the formula:

$$V = \frac{0.755\ (t + 460)}{P}\left(\frac{1 + W}{4{,}360}\right) \qquad (1)$$

where t = air temperature in degrees F.
P = absolute pressure in inches of mercury.
W = weight in grains of moisture per pound of dry air.

We find W for the entering condition of 100F and 100 lb gage pressure from the formula for vapor pressure e, where P = total pressure.

$$e = \frac{W\,P}{4{,}360 + W} \qquad (2)$$

By solving for W we find:

$$W = \frac{4{,}360e}{P - e} \qquad (3)$$

Where P is at 100 lb gage, then to change this to inches of mercury

$$(100 + 14.7 \times \frac{29.92}{14.7} = 233.45 \text{ inches Hg}$$

The vapor pressure, e, at 100F is 1.932 inches Hg. (See Psychrometric Table, H & V Data Sheet No. 87)

$$\text{Therefore, } W = \frac{4{,}360 \times 1.932}{233.45 - 1.932} = 36.4 \text{ gr per lb.}$$

To find V, we now substitute W and t in Eq (1)

$$V = \frac{0.755\ (100 + 460)}{233.45}\left(\frac{1 + 36.4}{4{,}360}\right)$$

$= 181\ (1 + .00835)$

$= 1.825$ cu ft of mixture containing one pound of dry air and 36.4 gr of moisture.

To determine air flow in pounds per hour, we substitute the known capacity of the compressor in the following equation:

$$\frac{1{,}500 \text{ cfm} \times 60 \text{ minutes}}{1.825 \text{ cu ft per lb}} = 49{,}315 \text{ lb per hr of dry air.}$$

To find the total heat to be removed by the refrigeration system, we determine the sum of the sensible and latent heat content of the air at both temperatures and multiply these differences in Btu per pound by the pounds of air being cooled per hour.

From this same data sheet, we can find h_g, the enthalpy of water at saturation, and h_a, the enthalpy of dry air. We shall use h_w as the symbol for the enthalpy of the water vapor content and h_t as the enthalpy or total heat of the mixture.

At 100F: W = 36.4; h_a = 24.0, and h_g = 1,104 Btu per lb of saturated vapor.

$$h_w = \frac{36.4}{7{,}000 \text{ gr/lb}} \times 1{,}104 = 5.725 \text{ Btu per lb of mixture}$$

h_a = 24.0 Btu per lb of dry air

h_t = 29.725 Btu per lb (enthalpy or total heat of the mixture)

At 55F: The value of e = 0.4356. The pressure is still 100 lb gage (233.45 absolute pressure in inches of mercury)

Using this new *e* value in the formula

$$W = \frac{4{,}360e}{P - e}$$

$$W = \frac{4{,}360 \times 0.4356}{233.45 - 0.4356} = 8.14$$

From H & V Data Sheet No. 87

h_a = 13.2 and h_g = 1,084

$$h_w = \frac{8.14}{7{,}000} \times 1{,}084 = 1.259 \text{ Btu per lb.}$$

h_a = 13.2 Btu per lb of dry air.

H_t = 14.459 Btu per lb.

Total heat to be removed equals

(29.725 — 14.459) × 49,315 = 742,646 Btu per hr.

The sensible heat to be removed can be determined in a similar manner by subtracting the value of h_a at 55F from h_a at 100F, and multiplying by 49,315 for this example:

(24.0 — 13.2) × 49,315 = 528,502 Btu per hr.

The balance or 214,144 Btu is the latent removal.

While the above calculations do not take into account the subcooling of the water after condensation, they are sufficiently accurate for normal industrial practice.

To find out the actual water that is removed from the compressed air by the mechanical cooling system, we subtract the grains of moisture per pound of dry air at the leaving condition from the grains at the entering condition and multiply the result by the pounds of dry air per hour at 100 lb pressure being handled by the compressor. This value divided by 7,000 grains per pound gives us our answer in pounds of water condensed out of the air.

Entering grains = 36.4

Leaving grains = 8.14

Grains removed = 28.26

$$\frac{28.26 \text{ gr} \times 49{,}315}{7{,}000} \text{ lb per hr dry air} =$$

199.1 lb per hr water removed, or 24.4 gal per hr.

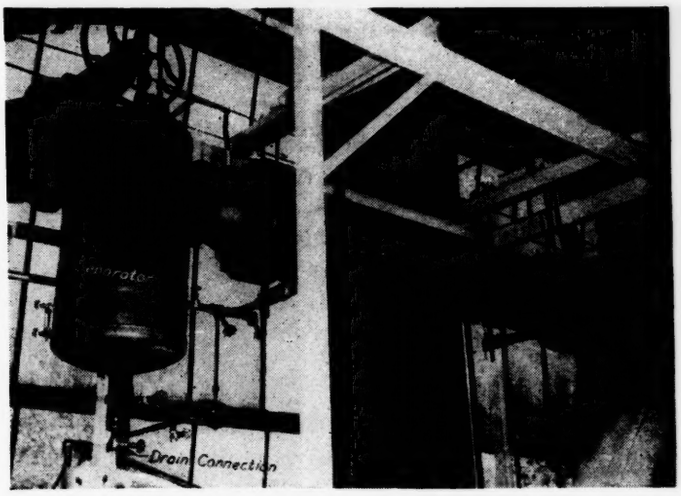

Fig. 2. Another installation showing the refrigerated aftercooler, separator and a part of the synchronous motor driving the air compressor.

From this, it is readily seen why mechanical refrigeration is necessary if dry air is to be obtained, as well as the amount of water that could be dropped out in the underground piping.

The next step in the solution of our problem becomes one of translating the load figures already calculated into terms of equipment necessary and horsepower required to accomplish the moisture removal.

In order not to create any erroneous ideas as to the magnitude of the refrigeration equipment required, we find that the free air volume of almost 11,000 cfm at 95F is enough to meet the compressed air requirements of most large industrial plants. As a matter of fact, this compressed air capacity is adequately serving the needs of a plant turning out approximately one million domestic refrigerators per year, plus a substantial number of diesel-electric locomotives. Smaller air volumes will require proportionately smaller refrigeration equipment, as will lower pressure delivered air.

While there are several ways of introducing cold surface in the path of the compressed air to accomplish moisture removal, many of these involve the design and fabrication of special heat exchangers. The pressure involved makes such special equipment quite expensive. It is, therefore, simpler and less expensive to utilize available standard equipment and to circulate chilled water through a combination aftercooler and separator. This combination completes the work done by the existing city water-cooled aftercooler. Operating expense and water usage can usually be kept to their previous values by using the exhaust cooling water from the existing aftercooler in the condenser of the refrigeration machine. If intercooler, jacket, and aftercooler cooling is being accomplished by water from a cooling tower, this reuse of water will not be possible. Should there be spare capacity in the tower, 3 gpm per ton of refrigeration will be the additional demand on the tower.

Condensing Equipment

The total load, in our example, of 742,636 Btu per hr can be handled comfortably by the 60 hp condensing

unit connected to a two-circuit dry expansion water chiller shown in Fig. 1. This picture was taken before insulation was applied to show all the details of the piping.

To arrive at the size of condensing unit required, we first choose a temperature of 45F for the chilled water, which will give us a 55F leaving compressed air temperature. To obtain 45F chilled water, we must operate at a 35F suction temperature which is obtainable with a 60 hp condensing unit. The choice of aftercoolers was determined by the manufacturer of this equipment to meet the requirements of the particular load conditions. It will be noted that each compressed air line has its own aftercooler so that the available space could be utilized to the best advantage and still provide adequate access to the refrigeration equipment.

The 60 hp condensing unit can handle nicely the load of 742,636 Btu per hr, since this is the maximum possible load under the conditions given in our example. It is a well known fact that good practice dictates some safety factor be provided when installing air compressors. In this particular case the safety factor was figured to be of the order of 10 to 15% based on the compressed air consumption figures which had been collected for this particular plant. In sizing your load, such figures are extremely helpful as a guide, but they must be weighed carefully in the light of future plans for plant output which may entail the use of more compressed air than in the past.

The refrigerant piping is provided with heat exchangers, flexible metal hose, dryers, liquid line strainer and solenoid valve, in accordance with good industrial refrigeration practice.

Fig. 2 is another installation where a single aftercooler could be used because of the extreme height available in the compressor room.

No unusual problems were presented in the refrigerant piping or equipment mounting of either of these installations. The chilled water circuits were straight forward and simple. The only problem was one of control, and that was divided into two parts—safety and operation.

Controls

Since automatic operation was desired, a combination of electrical and mechanical controls was used to protect against various kinds of operational failures. The condensing unit starter was electrically interlocked with the chilled water pump and, in addition, there was a pressure switch in the chilled water pump discharge line to protect against any stoppage of the flow of water to the chiller. The operating thermostat and the low limit thermostat in the chilled water circuit were backed up by a differential pressure switch connected across each end of the chiller in the water circuit. In the event of failure by both thermostats, the pressure switch will shut down the condensing unit when the formation of slush ice in the chiller changes the pressure drop across the water side of the chiller. A pressure switch in the condenser cooling water line operates in the event of water failure. This device is supplemented by the high pressure switch in the refrigerant circuit to shut down the condensing unit when the head pressure gets too high for any reason. A liquid line solenoid is used to permit operating the system on a pump-down cycle to preclude any possibility of liquid flood-back with the resulting consequent breakage.

The problem of operational control arose because of the variations in demand for compressed air due to two-shift operation of the plant as well as the seasonal variation in cooling water temperature. In the system shown in Fig. 4, operation at capacities of 100, 75, 50, 37½, and 25% were made possible by a combination of a two-speed motor and two-cylinder by-pass solenoid valves. Dual back pressure switches actuated the by-pass valves. In order to keep down operating cost, half-capacity operation was achieved through half-speed operation of the motor by means of a selector control and timer actuated by chilled water temperature. Once every minute the selector has the opportunity to determine whether high or low speed is required and, by means of relays, satisfy the need. The required speed is maintained until at some subsequent one minute interval, a change is called for. Capacity requirements of 75, 37½, or 25% are met by solenoid operation, without regard to any time cycle, in response to the back pressure switches.

Ordinarily, such complicated operational controls are not necessary as 50% capacity reduction usually suffices. This can easily be provided for the system illustrated in Fig. 1 by the addition of one more liquid line solenoid valve and another solenoid to by-pass 50% of the cylinders of the compressor. A drop in compressed air requirements, resulting in a shut-down of one air compressor, could automatically close the liquid line solenoid in the chiller circuit serving the aftercooler on the appropriate air line. A decrease in cooling water temperature, causing a drop in the mechanical refrigeration load on both aftercoolers, could be handled by means of the by-pass solenoid under direction of the auxiliary back pressure control switch.

Occasionally, the necessity will arise for extremely dry compressed air for special process work. Within reasonable limits, this can be accomplished by the same general method employing cold brine instead of chilled water in the aftercooler. Dual aftercoolers, operated one at a time with adequate time for defrosting between change-overs, will be necessary to prevent passing snow into the air lines and defeating the purpose of the installation. The initial expense for adequate controls will be amply repaid over the life of the installation. It can further be demonstrated that the cost of this conditioned air can be quickly amortized. It should result in large dividends to the purchaser.

This is especially true where the use of dry air is required for product quality improvement, as in the case of a refrigerator motor assembly. Here the cotton stator windings must not be permitted to regain moisture after having been dried out in autoclaves prior to assembly in the hermetic compressor.

While moisture in compressed air is a menace, it can be tamed by means of industrial refrigeration. Moreover, in the process of taming this menace, savings can be effected which will amortize the cost of the investment in a short time.

Design of Push-Pull Exhaust Systems

JOHN F. EGE* and LESLIE SILVERMAN†

Preliminary data are given on design of push-pull ventilation for tanks. The value of jet confinement is clearly demonstrated. Basic characteristics, velocity contours and entrainment of such jets are presented. Liquid level influence on velocity contours is shown in a series of charts. Typical examples indicate the merits of the confined jet with respect to the free jet.

PUSH-PULL or blow exhaust systems, sometimes denoted as air-curtain procedures, have been used to a limited extent in industry for the control of open surface tank operations. Occasionally this method has proved successful in certain other operations such as bagging-dust control. In contrast with a conventional exhaust system, this method results in lower air flow requirements and, as a consequence, savings in heat losses, ventilation equipment and ductwork.

There are many practical objections to the use of push-pull exhaust systems, particularly where access to the controlled area is required at frequent intervals. This objection is obvious, since excessive contamination results. Interference of solvent or acid coated parts or objects with the jet provides a new mechanism for atmospheric contamination in the ambient atmosphere. Irregular surface cooling and heat loss are other limitations. There are, however, many operations conducted over long periods in wide tanks where ventilation requirements of exhaust slots or overhead canopies become uneconomical because of the large air quantities involved. Push-pull ventilation may be of value in these instances.

Information in the literature on design of push-pull exhaust systems is limited. Design values for one type are given in the patented air-curtain data shown in the 1938 American Foundrymen's Association code. A practical method of achieving push-pull exhaust systems is shown by Malin (1)**. In this case, the push or supply is produced by a pipe drilled with a number of holes and supplied with compressed air. The total exhaust volume required per square foot of tank with this additional aid is approximately half that for suction only. Malin recommends that the air be supplied by 1½- to 2-inch steel pipe with 5/32 to 3/16 inch diameter holes spaced at intervals of 2 to 2½ inches and discharging streams across the tank at an angle of approximately 5 deg below the horizontal. Air pressure should be about ¼ lb per sq in. or 7 inches of water. Under these conditions, the air velocity at the holes is about 9,000 fpm and carries for 5 ft. This air jet requires approximately 8 cfm per foot of tank length and it is recommended that a separate air supply be used for this purpose. This can be furnished from a compressed air line or a blower. These low rates have been found satisfactory on one installation in the authors' experience for removing steam vapors with moderate dispersion.

Battista (2) has shown results of push-pull with a similar arrangement but using the semi-lateral exhaust hood. He employed a blow pipe with 1/16 inch holes and 1½-inch spacing. For a 3½ ft wide tank, 1½ cfm per foot of tank is recommended for the blow pipe (9½-inch static pressure) with heavy contaminant dispersion and 1 cfm per foot (4 inches s.p.) for light contamination. This results in a ventilation of 75 cfm per sq ft of tank with effective control as compared to 150 cfm per sq ft without the blow pipe. Push-pull apparently halved air flow requirements for this particular case.

No appreciable information is available on the nature of jet air flow over tanks or on actual amount of entrainment produced by such jets. Considerable data (3 to 9) on jets in space have been presented by aerodynamics investigators, and work on supply to rooms or enclosed spaces by ventilation research workers. This information is based on the characteristics of unconfined or partially confined jets and is not directly applicable to flow over tanks and vats.

Data reported in this study were obtained in this laboratory on an experimental apparatus and consist of information on the freely expanding jet and one confined on both sides and below at various levels. In addition, flow requirements for both primary and entrained air are presented for various positions of the bottom plane. It should be recognized that these data are preliminary and that further work is planned.

Procedure

In these experiments a ¾- x 12-inch slot was used, connected to a 4½-inch duct by means of a transformation piece. The slot face was oriented in a vertical plane with the air flow directed in a horizontal plane. Velocity across the slot was maintained at 4,000 fpm by means of a damper and a metering orifice. A felt filter cloth section was provided in the duct to produce a uniform discharge distribution. The maximum variation in the face velocity from one end of the slot to the other was less than 15%.

The following instruments were used for velocity measurements: a standard pitot tube (Ower or modified NPL) with vertical and inclined water and alcohol

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**Numbers in parentheses refer to bibliography at the end of the article, unless otherwise indicated.

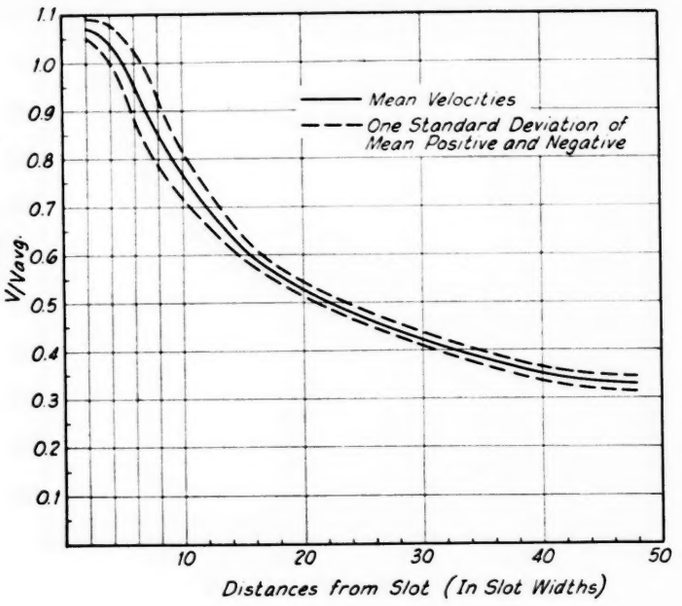


Fig. 1. Velocity-distance relationship.

gages; an impact pressure unit of 7 hypodermic needles (No. 19) 3½ inches long, bent at right angles with a ½ inch probe spaced two inches on centers and rigidly mounted; a Velometer. To facilitate measurements in the case of the freely expanding jet, a positioning jig was used with the hypodermic needle impact tubes in order to vary distances out from the slot face and distances above and below the slot centerline. The pitot tube and a Velometer were used in confined jet studies.

Airflow Patterns – Free Jet

For simplification, velocities in space measured by 7 impact tubes are expressed as a ratio of the mean velocity in space divided by the mean slot face velocity. These ratios are plotted against distances, in slot widths, from the slot face. Results of such measurements are plotted in Fig. 1.

The standard deviation is a measure of the reliability of the measurements and the value is considered as a plus or minus one from the mean. The dotted lines of Fig. 1 thus show the range over which the data is within ± 3% error over-ail. Original data are given in Table 1, with all distances expressed in inches rather than slot widths. Velocities, however, are expressed as ratios of the measured value to mean face velocity. Mean velocities and standard deviation of the mean are included.

By conventional analysis, the following equations fit these data:

For the range, 0 to 2 slot widths

$$V/V_{avg} = 1.07 \qquad (1)$$

For the range, 6 to 16 slot widths

$$V/V_{avg} = 1.612 - 0.85 \log W \qquad (2)$$

For the range 16 to 48 slot widths

$$V/V_{avg} = \frac{6.6}{W} + 0.20 \qquad (3)$$

where V = velocity in space, fpm

V_{avg} = average slot velocity, fpm

W = distance in space from slot face in slot widths

Certain differences from McElroy's data on discharge phases for a 10-inch pipe in a mine drift (3) exist. The phase of constant velocity ratio is relatively short and extends to only two widths. The "transition phase" extends from 2 to 6 widths. The logarithmic portion of the curve follows rather the second phase (McElroy) in which the velocity ratios vary inversely as the square root of the distance. The next phase, in which the velocity ratios vary inversely as the distance from the slot face, corresponds to his third phase. No fourth phase was observed.

Expansion Angle—The tangent of the angle of expansion corresponds to 22.6° and is in good agreement with values of 24.2° presented by Tollmein (4), 24.5° by Ruden (5), 18° by Forthmann (6), and 20.5 to 26° observed by Tuve and Priester (7). Kuethe (8) has also presented data which is in accordance with these observations.

Static Pressure in Space—Within one to two widths of the slot face, measured static pressure readings were positive. These values were 0.15 inch of water or approximately 15% of the velocity pressure. Readings further away from the slot passed through zero, then became negative at approximately 5.8% of the velocity pressure.

Airflow Pattern – Confined Jet

The confined slot is shown in Fig. 2. In these studies, the horizontal or bottom plane (tank liquid

TABLE 1—RELATIVE VELOCITIES* IN SPACE

D**	Distance along Slot Length, Inches							Mean	S†
	0	2	4	6	8	10	12		
2	1.04	1.08	1.09	1.05	1.09	1.07	1.06	1.07	0.020
4	0.96	1.01	1.10	1.04	1.07	1.06	1.02	1.04	0.036
6	0.86	0.94	1.06	0.93	0.95	0.99	0.85	0.95	0.074
8	0.78	0.86	0.96	0.85	0.85	0.90	0.76	0.85	0.068
10	0.71	0.79	0.83	0.77	0.78	0.81	0.69	0.77	0.050
12	0.66	0.73	0.72	0.71	0.71	0.72	0.64	0.70	0.035
14	0.61	0.67	0.65	0.65	0.66	0.61	0.60	0.64	0.030
16	0.58	0.62	0.59	0.60	0.61	0.56	0.58	0.59	0.020
20	0.52	0.56	0.51	0.53	0.54	0.53	0.51	0.53	0.012
24	0.48	0.51	0.46	0.47	0.49	0.48	0.47	0.48	0.016
28	0.44	0.46	0.42	0.42	0.45	0.44	0.43	0.44	0.015
32	0.40	0.43	0.40	0.38	0.41	0.41	0.41	0.41	0.016
40	0.35	0.36	0.36	0.32	0.36	0.36	0.35	0.35	0.014
48	0.32	0.33	0.36	0.31	0.33	0.34	0.33	0.33	0.016

*Expressed as ratios of measured value to mean face velocity.
**Distance out from slot face, inches. Slot width ¾ inch.
†Standard deviation of mean.

level) was raised and lowered to measure the effect of position on the airflow pattern. As the level fell, the air jet dipped sharply toward the bottom plane, at approximately a 30° angle from the plane of the unconfined jet. This effect was observed with a 4,000 fpm slot velocity and also at lower slot velocities. As indicated in Fig. 2, at the slot end, airflow under the jet created a circulating vortex. At the opposite end, highest velocities were those adjacent to the plane and lowest next to the side walls.

Effect of Position of Bottom Plane in Confined Jets—To determine the influence of the bottom plane location in confined jets relative to the slot, several traverses were made at several positions of the bottom plane. Data from these tests are plotted in Fig. 3 to 8. In these curves, dotted lines represent a reversed airflow. In all cases, the velocity ratios shown represent the horizontal component, as no attempt was made to align the pitot tube with the resultant vector or true air direction.

Fig. 3 to 8 are 3-plane diagrams and selection of values from these curves is based on use of the three plane dimensions. The curves are velocity ratios at stated distances in slot widths from the hood face. The major variable between Fig. 3 to 8 is the distance of the plane below the slot centerline. Thus in Fig. 3, since this distance is so small, values of the individual curves just intercept and extend slightly to the left. In succeeding figures a greater portion of the curve extends to the left side.

To utilize these curves, the number of slot widths from the face fixes the distance on the right hand axis. From this point the value of the number of slot widths below or above the slot centerline is selected on a line parallel to the left-hand axis. From this locus a vertical line intercepting the velocity ratio curve fixes the velocity ratio value read from the vertical axis by moving in a plane parallel to the left- and right-hand axes.

As an example on use of these charts, assume that only measurements at 40 slot widths are employed and that a liquid level relative to slot centerline is to be maintained at 3.8 widths below slot centerline. The values must then lie between Fig. 4 and 5. Referring to Fig. 4 at 40 slot widths out from face, the nearest measured point to 3.8 is the furthest left or 0.41 for $\frac{V}{V_{avg}}$ at approximately 1.5 slot widths (plane 2.67 widths). In Fig. 5 the same value is 0.41 at 3.8 slot widths (plane 5.32 widths), hence for practical purposes it may be assumed that if a plane were established at 3.8 slot widths below the slot, the same value (0.41) would apply.

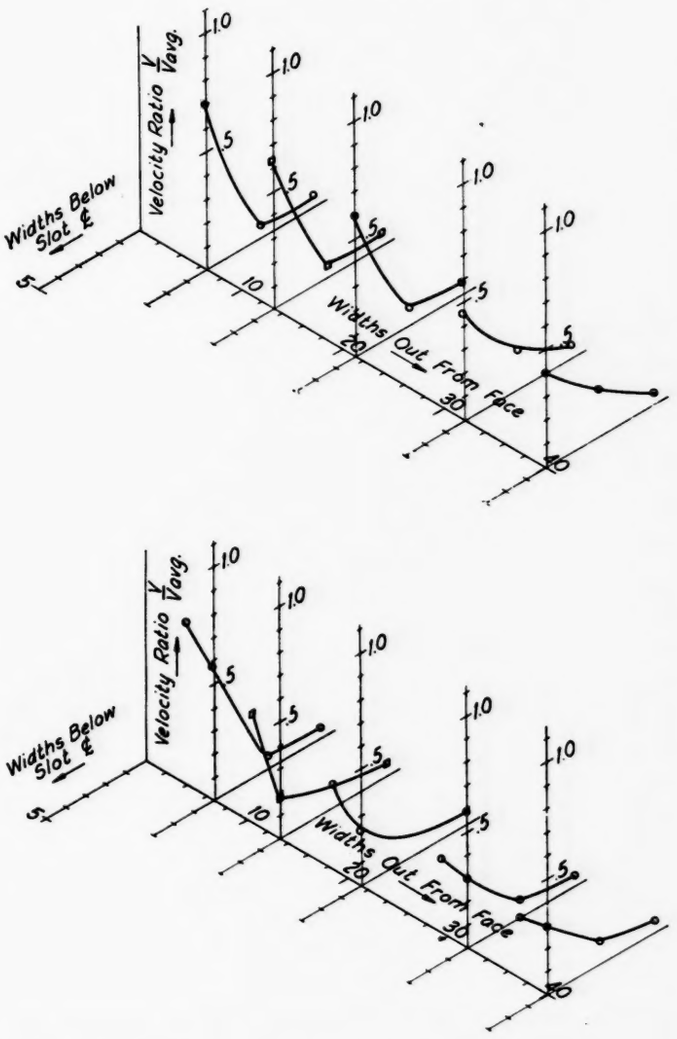


Fig. 3 (upper). Velocity contours for confined jet tank. Bottom plane at 1.33 widths below slot. Fig. 4 (lower). Bottom plane at 2.7 widths below slot.

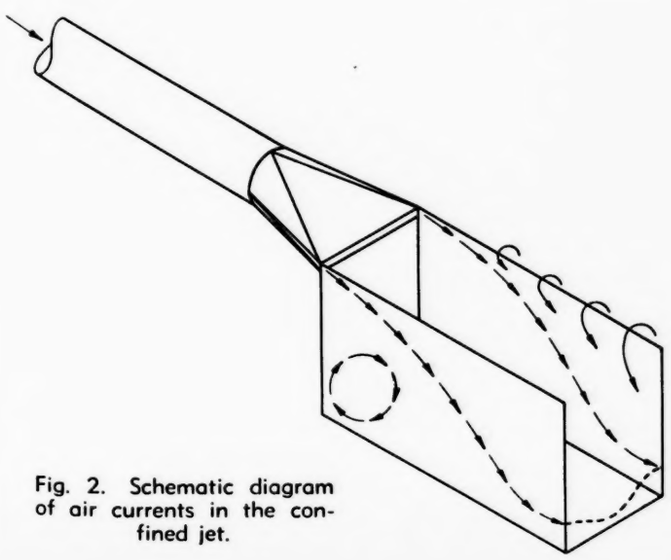


Fig. 2. Schematic diagram of air currents in the confined jet.

It can be noted in confined jets that there is no spillage of air out of the top surface; but, rather, air is drawn in by entrainment. It was also found in the majority of cases, that maximum velocity ratios of the confined jet are greater than those for the corresponding free jet at distances from the slot greater than 22 widths.

Entrained Air Measurements for the Confined Jet—Airflow was measured across the open end, as illustrated in Fig. 2, by vertical and horizontal velocity traverses. Measurements were made to include two inches (2.67 slot widths) above the slot centerline. Values were converted to average velocities through the open end. The total airflow thus obtained includes primary and entrained air. These results are summarized in Fig. 9 where entrainment ratio (ratio of

entrained primary air) is plotted with respect to floor level position expressed in slot widths below the slot centerline. In this figure, the floor level is designated "liquid level," as in practice.

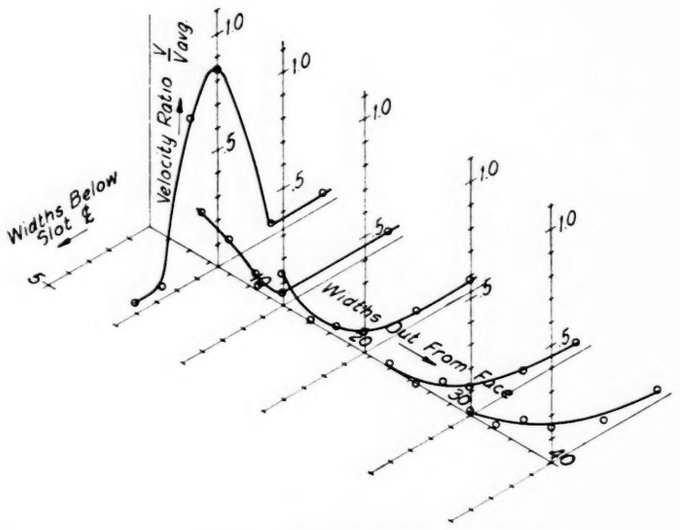


Fig. 5. Velocity contours for confined jet tank. Bottom plane at 5.3 widths below slot.

It should be emphasized that these measurements were made at 40 widths from the slot face. The equation for the best fit line over the range in Fig. 9 is

$$E.\ R. = 0.55 + 0.12\ D \qquad (4)$$

where E. R. is the entrainment ratio (ratio of entrained to supply air volume)

D = location of liquid level in terms of slot widths below the slot centerline.

In tank ventilation, two push-pull methods of ventilation can be used utilizing either free jets, or jets confined on three sides. Data for both types were presented. Both types require secondary exhaust hoods. The free jet has practical difficulties which will be discussed. The following sample calculations are presented for the confined jet of the type shown in Fig. 2.

Confined Jet Example

Ventilation is to be provided for a tank 4 ft x 15 ft of the type shown in Fig. 2 with a minimum control velocity of 75 fpm. The tank is filled to within approximately 4 inches of the slot (liquid level).

Since design information obtained from this investigation is based on measurements at 40 slot widths, the slot width can be determined by:

$$W_S = \frac{4 \times 12}{40} = 1.2 \text{ inches}$$

Liquid level relative to slot centerline is therefore

$$4 \text{ inches} + 1/2\ W_s = 4 + 0.6 = 4.6 \text{ inches}$$

$$\text{or } \frac{4.6}{1.2} = 3.8 \text{ slot widths below slot centerline}$$

Velocity ratios therefore lie between the values obtained by interpolation from Fig. 4 and 5. These are essentially the same at 40 widths

$$V/V_{avg} = 0.41$$

Therefore the supply quantity equals:

$$Q_S = W_S L_S V_{avg} = \frac{1.2}{12} \times \frac{15 \times 75}{0.41} = 275 \text{ cfm}$$

$$\text{and } \frac{Q_s}{L} = \frac{275}{15} = 18.3 \text{ cfm per ft tank length}$$

From Fig. 9, the entrainment ratio is approximately one, and therefore the entrained volume

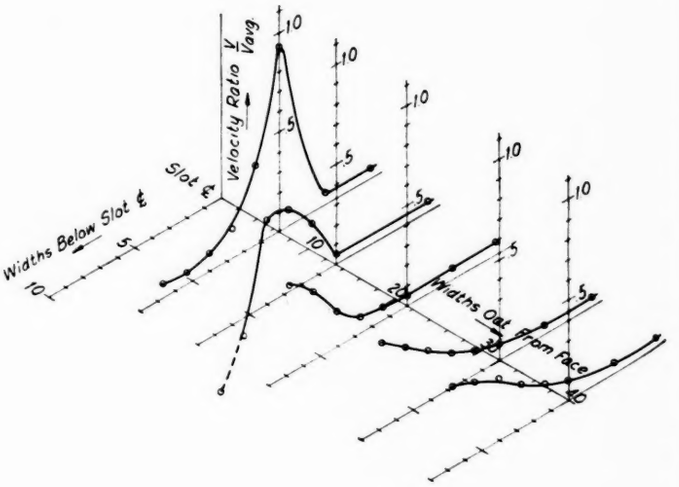
$$Q_E = (E.R.)\ (Q_S) = (1.0)\ (275) = 275 \text{ cfm}$$

The total exhaust volume is then

$$Q_T = Q_E + Q_S = 275 + 275 = 550 \text{ cfm}$$

Total air to be moved by blowers = 275 + 550 = 825 cfm

Since the liquid level is to be maintained 4 inches below the slot we allow a safety factor for spillage of 2 inches above the centerline of the slot. Therefore,



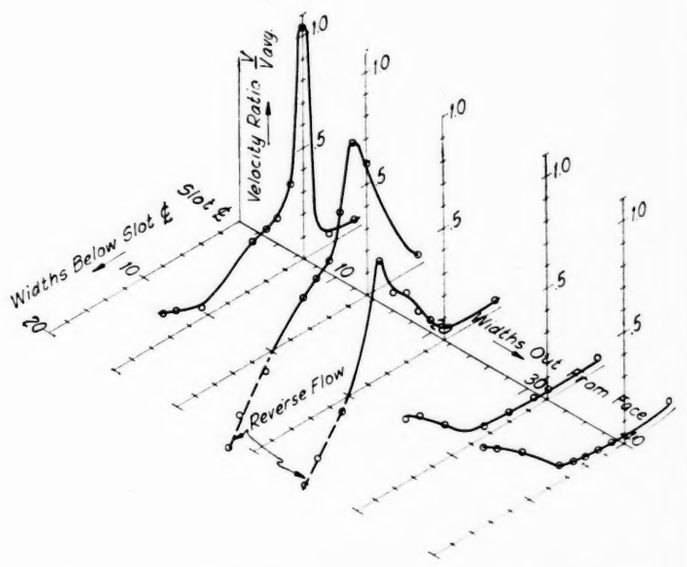


Fig. 6 (upper). Velocity contours for confined jet tank. Bottom plane at 8.0 widths below slot. Fig. 7 (lower). Bottom plane at 17.3 widths below slot.

$4 + 2 + 1.2 = 7.2$. We arrive at an approximate height of 7 inches for the secondary exhaust hood. It is preferable, but not always possible, to locate this hood close to the liquid level in order to minimize "air splash out." A slight pitching upward of the supply jet facilitates the location of the exhaust slot at tank level only if air can be admitted below the slot, as mentioned later. Fig. 10 shows a possible arrangement for this confined jet which amounts to a complete push-pull hood component for the tank. Air is exhausted at a rate of 550 cfm. On a basis of tank area, the total ventilation rate is approximately 14 cfm, including total air moved by blowers.

This air flow per unit area of tank may seem abnormally low compared to values computed from other push-pull design data but it should be remembered that this jet is continuous and reinforced on three sides as compared to those types using the jet above the tank. It should also be mentioned that these calculations do not consider the reinforcing effect of the exhaust slot suction on the discharge velocity at points of coincidence, which is an additional safety factor.

If the primary or supply air is not heated, the heated air ventilation requirement is only the entrained air or 5 cfm per square foot of tank surface. In actual practice, however, cases arise in which the air curtain method does not result in savings of heat of this magnitude. This situation arises in particular when the tank liquid is heated.

Since the push-pull method results in higher velocities at the surface of the liquid, greater evaporation and heat losses can occur. For water, the following formulas can be used to estimate losses for parallel flow to a horizontal water surface:

$$\text{Heat loss} = H = 97A\left(1 + \frac{V}{230}\right)(P_w - P_a)$$

Btu per hr (5)

$$\text{Moisture loss} = W = 0.093A\left(1 + \frac{V}{230}\right) \times$$

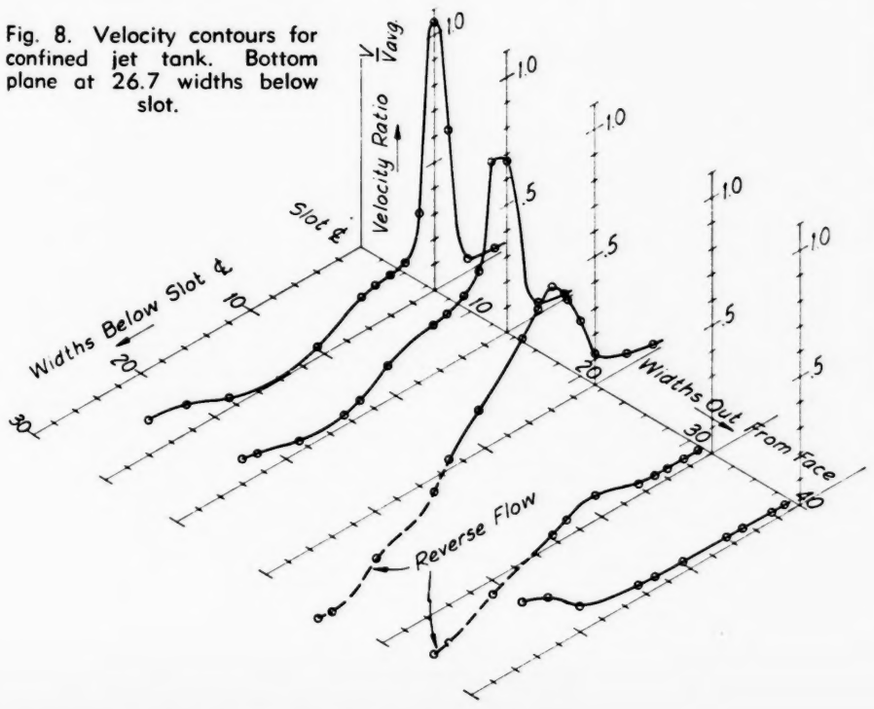


Fig. 8. Velocity contours for confined jet tank. Bottom plane at 26.7 widths below slot.

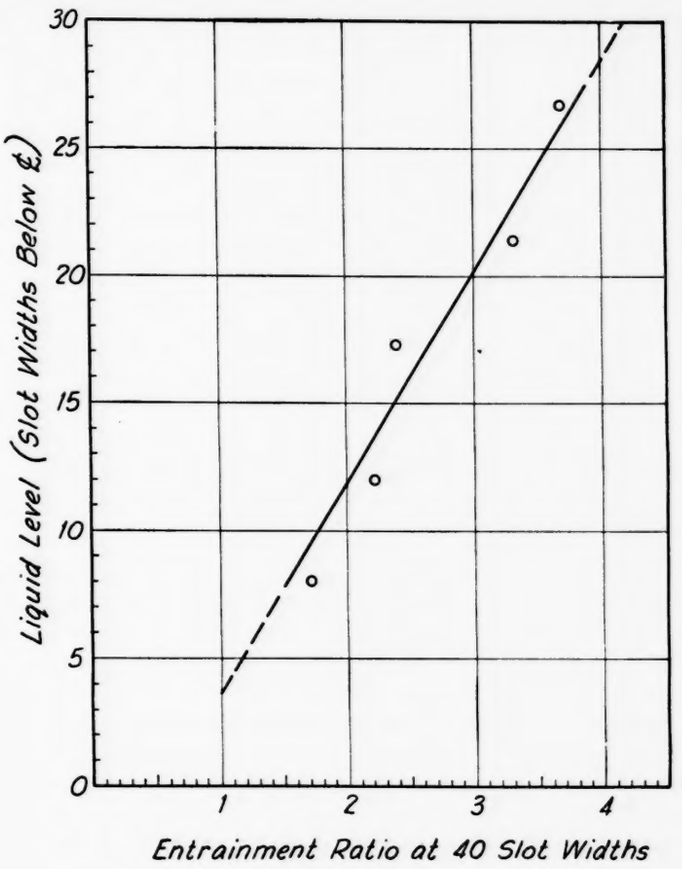


Fig. 9. Entrainment ratio at 40 slot widths for various liquid levels.

$$(P_w - P_a) \text{ lb per hr} \qquad (6)$$

A = tank area in square feet
V = velocity in fpm (standard conditions)
P_w, P_a = vapor pressure over tank and in air, respectively (in inches of mercury)

In the problem given, this additional heat loss, when added to that due to the airflow, is lower than for a conventional exhaust slot because total ventilation per square foot of surface is considerably reduced by use of a "free" rather than a confined jet. In the case of volatile solvents, high surface evaporation cannot be tolerated. Salt or acid solutions may depress the rate of evaporation.

Another practical difficulty encountered is obtaining a low uniform slot velocity. The restriction shown in Fig. 10 can be used; another approach is to provide two perforated sheets along the slot length arranged so that one is fixed and the other slides on top to constitute an adjustable damper. This latter device should be located in the supply duct some distance from the actual slot.

Free Jet Example

In the case of the low velocity free continuous jet, there are further practical limitations. One restriction is the fact that the slot ductwork should be designed so that air enters on the under side of

the slot. This prevents downward deflection of the jet and increased surface losses. A low uniform slot velocity must also be provided. These factors require special ductwork which has no material advantages over the multiple holed high velocity pipe jets mentioned previously.

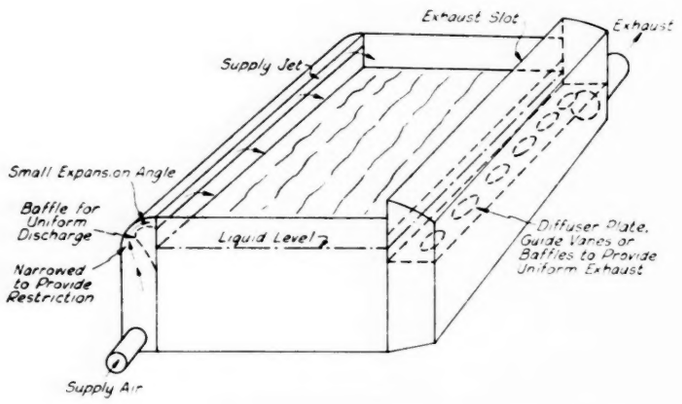


Fig. 10. Push-pull exhaust component for tank utilizing confined jet principle.

A tank similar to the one used in the example for confined jet calculations (4 × 15 ft with 75 fpm control velocity) is to be designed. Jet air requirements are obtained from Fig. 1. Obviously, this jet should be directed upward at an angle of 11 to 13°, so that the air blast clears the tank along its entire width.

Slot width calculation is the same as in the confined jet and will be based on measurements at 40 slot widths, hence $W_S = 1.2$ inches. The jet should be located above the tank, if possible, to allow air to enter below. Liquid level is maintained, in this case, at 4 inches from the tank top.

The supply volume is computed from Fig. 1 and the slot width viz:

$$\frac{V}{V_{avg}} = 0.35$$

$$Q_S = \frac{1.2}{12} \times \frac{15 \times 75}{0.35} = 322 \text{ cfm}$$

$$Q_{S/L} = \frac{322}{15} = 21.5 \text{ cfm per ft}$$

From Tuve et al (9) an approximate entrainment ratio is obtained based on their data for a 24 inch x 1 inch slot. This value for E.R. is approximately 6.5.

$Q_E = \text{E.R.}\ (Q_S) = 6.5 \times 322 = 2{,}090$ cfm

The total volume to be exhausted

$Q_T = Q_S + Q_E = 322 + 2{,}090 = 2{,}412$ cfm

Total air to be moved by blowers $= 322 + 2{,}412 = 2{,}734$ cfm

Or per square foot of tank surface, total ventilation

$$\frac{Q}{A} = \frac{2{,}734}{60} = 45.6 \text{ cfm per sq ft}$$

It can be seen that this requirement is intermediate between the confined jet and suction alone.

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Basic Facts Concerning Pennsylvania Anthracite

More than 99% of the anthracite produced in North and South America originates in portions of nine counties in northeastern Pennsylvania. The surface area of the deposits is 484 square miles. Reserves of Pennsylvania anthracite are officially estimated to be between 160 and 170 years at present rates of extraction and under present methods of mining.

Mining is of two types—underground, approximately 74% of the total, and surface or stripping, the remaining 26%. Underground mining varies in depth and some operations are as deep as 1,250 ft below the surface, or 500 ft below sea level.

Anthracite heats 5,000,000 homes and buildings and is the predominant domestic heating fuel in the New England and Middle Atlantic States and in the Provinces of Ontario and Quebec. It is also extensively used in certain lake state areas which can be economically reached by water carriage over lake docks.

Heating and Ventilating Schools with Unit Ventilators

ALBERT J. NESBITT
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This is the fourth of a group of articles dealing with suitable heating systems for school buildings. Each article by a different engineer-specialist covers a specific type of heating system satisfactory for school buildings.

THE unit ventilator system of heating and ventilating possesses certain inherent characteristics that make it especially well suited to the peculiar and particular requirements of school building heating and ventilating. That the unit system has these characteristics to a greater extent than any other method of heating and ventilating buildings of this type is widely accepted by school building authorities who have a true understanding of the nature of the requirements.

Problems of schoolroom heating and ventilating are not solved by simply providing adequate heating capacity under design conditions. The real test of the adequacy of a system is its ability to meet the varying day-by-day demands of the classroom.

The peculiarities to which reference was made are many and varied. The major one is the need for cooling, even when the outdoor temperature may be 40F or lower. During periods of occupancy, cooling is required for more hours than heating. One or more rooms of the building frequently need cooling while other rooms in the same structure may require heating. Therefore, room-by-room treatment is required, with outdoor air being supplied at the temperature and quantity needed for the individual room, without the influence of the particular requirements of some other room in the same structure.

System Cuts Heating Needs

Not only are the heating needs of the individual rooms intermittent, but since the schoolrooms are generally used only during part of the daytime hours, this system is out of use more hours than in use during the heating season.

Considerations of economy dictate that a schoolhouse heating and ventilating systems should be chosen, among other things, for its capacity to quickly heat the building after long shut-down periods. Since the unit system warms each room by recirculation, and does not draw in any outdoor air until the room is up to temperature, the normal heating capacity of the system is more than doubled; the time required to bring rooms up to occupancy temperature, compared with many other systems, is about half.

A school building is not a single purpose structure. In many communities it serves many purposes—indeed, one measure of its usefulness is its capacity to function in that way. Some parts of the building frequently continue to be used long after "school is out." Therefore, considerations of economy suggest that the system must be flexible. Individual unit ventilators in each room make it possible to control the number of rooms to be conditioned for the useful occupancy that is required. No more or no less rooms need be conditioned than are to be used.

Air Requirements

The density of occupancy makes necessary the constant supply of a minimum quantity of outdoor air. Most authorities are in agreement that 10 cu ft of air, when measured at 70 F and 29.9 deg barometric pressure, should be supplied every minute for every person in the building.

There are times when this outdoor air rate, which we will refer to here as the ventilation rate, is both above and below the quantity of outdoor air required for heat removal. Let me illustrate. Experience clearly proves that a fully occupied classroom into which the sun is shining, will require more than 10 cfm of outdoor air per person on most days when the outdoor temperature is not lower than 40F. Still another room in the same building which is not getting the heat of the sun will require only outdoor air equal to the ventilation rate. In some rooms as much as 20 cfm of standard air per pupil is required at times to prevent overheating. This is double the accepted or normal ventilation rate. Thus we have a situation in which rooms side by side may require as little as 10 cfm of outdoor air per person, or as much as 20 cfm of standard outdoor air per person.

Fuel Requirements Reduced

Many systems, where all the air is handled by one or more fans supplying all or several sections of the building, must permit the maximum outdoor air requirement to become the minimum. In this procedure, the fuel requirements for ventilation are approximately double that of a unit system. It is well known that the unit ventilator system operates under a thermal system of ventilation control in which no outdoor air in introduced into the room until the room has been warmed to the proper temperature. This has been previously referred to as the recirculating process. It will bear repetition that on this cycle of control the unit operates as a unit heater. In doing so, it has upwards of 200%

of its normal heating capacity than when it is used for both heating and ventilating and, therefore, it has over 200% heating capacity in relation to the need.

When the room reaches the desired temperature, then, automatically, outdoor air is admitted at the ventilation rate only, during which period indoor and outdoor air are mixed by the unit. As the thermal demands of the room change, the unit automatically introduces larger quantities of outdoor air and recirculates less room air. However, this is only done when the demand is for air at lower than room temperature. Consequently, more air than the ventilation rate is supplied only after all sources of heat have been automatically shut off. Thus an economy is achieved that is unobtainable in any other system; moreover, a thermal environment is provided equal to the thermal environment of systems which handle a constant quantity of air without regard to the individual requirements of the room.

Comfortable Environment

A satisfactory schoolhouse heating and ventilating system must possess the capacity to remove heat. Certainly a measure of adequacy in a system for this purpose is its ability to provide an environment in which the total heat exchange between the body and it surroundings is at or near the normal metabolic rate. Here again, we have a peculiarity of the schoolhouse heating and ventilating requirement that is not common in many heating requirements. It is the large relationship of glass to the total outside wall surface. If these surfaces are permitted to drop too low in temperature, those persons near these surfaces will not be comfortable, even in the presence of a 70F room temperature.

Objectively stated, our problem is not to heat the building but to provide a satisfactory thermal environment for its occupants. True, this requires that the building be heated. However, more than that is needed if the criteria by which we are to measure the worth of a system is its ability to provide a satisfactory comfort condition for its occupants.

Comfort is related to the temperature of the surrounding walls and objects as well as to the air temperature. A satisfactory thermal environment is not provided by simply keeping the room temperature at or near 70F. You cannot rest the whole case for comfort on air temperature alone. This might be paraphrased by saying that you cannot rest the whole case for a satisfactory thermal environment on what the room thermostat indicates. The room thermometer does not possess the ability to measure comfort; it simply measures air temperatures.

Comfort is more than that, for when surrounding walls or glass temperatures are low, the human body radiates heat to these surfaces at too fast a rate for comfort. Therefore, in this particular field, the system must possess the capacity to relate the temperature of the air introduced into the room, not only to the room temperature but to the outdoor temperature as well. The minimum temperature at which air is introduced into the room must be elevated as the outside temperature drops. To achieve this, the heating should be done on a room-by-room basis. In circumstances of low outside temperatures, a protective blanket of warm air is needed between the cold glass surfaces and the room occupants. This blanket serves to raise the temperature of the cold surfaces.

The present-day unit ventilator, when automatically controlled, may be equipped to provide this needed protection from cold surfaces and to provide it on a basis that relates minimum temperature at which the air is supplied, not only to the room temperature, but the outdoor temperature as well.

First Costs Are Low

Competitive bids taken upon identical buildings for other systems in the past several years have shown the unit systems to be lower in first cost than any other controlled, mechanical heating and ventilating system, whether of the warm air or steam type. This is readily understandable when one considers the fact that the tool equipment with which each individual plant worker in a unit ventilator plant operates has a value upwards of $10,000 per man, whereas the tool equipment employed by mechanics working on a building fabricating a ventilation system with ducts and other needed materials, seldom has a value as much as $1,000 per man.

The inherent characteristics of the unit ventilator system obviously result in operating economies that cannot be achieved by any other present-day system of controlled mechanical ventilation. Maintenance costs on unit ventilators are practically nil and it is a fact that thousands of units that have been in operation since before 1920 are still in use, being the original equipment of the manufacturer without replacements.

A careful analysis of horsepower required will show that unit ventilator systems require 25% less horsepower than systems which blow air long distances against the resistance of air travel. These power savings frequently run as high as 50%.

The low first cost of this system, moreover, does not tell the whole story, since the elimination of fan and heater rooms, duct spaces and heating flues, result in lower general building costs which has been variously estimated at from 1% to as much as 6% of the construction cost.

The ever increasing demand on the school as a community center requires a school plant which must be flexible for the varied and intermittent uses of all rooms. The community center demands efficient and economical use of installed facilities. The need for flexibility is widely understood and accepted by the enlightened educator. In fact, school authorities are insisting upon structures that are fluid and equipped with systems that are capable of being changed with a minimum of inconvenience as changes take place in the educational program. The advantages of the unit ventilator system in a changing program of this kind are at once apparent. It provides the school plant with equipment that can be readily operated, unit by unit.

Our best assurance that time will not outmode the unit system in this everchanging world, is the units' adaptability to changing requirements.

Two Boilers May Require a

Third Means of Equalization

T. W. REYNOLDS

ONCE, a boiler safety loop was an unheard of thing. Later, when the advisability of its use under certain conditions dawned, we were quite apt to use one, but still only one, regardless of whether there were more than one boiler. Such a practice is wrong. Boiler water may flow through a common loop from one boiler to the other, a fact that is not always appreciated. In more recent years we have progressed to a knowledge that two steel boilers, each of two-piece construction, Fig. 1, require an additional or third means of equalization when piped together.

The two-piece boiler was designed about 38 years ago. Its circulating connections, Fig. 2, provide the only passage for water flow from firebox to shell, hence it is in effect two boilers in one. Most of the water is up in the shell, and the shell and firebox have different zones of heating surface and steaming rates. The shell contains the indirect heating surface, but the firebox has a lot of direct heating surface with a heat absorption value ten times that of the indirect. This valuable direct surface is quickly chilled by opening of fire doors, but regardless of this the boiler may reverse its flow at different ratings. Where there are two boilers interconnected, one may at times circulate in a reverse direction while the other circulates in the proper direction, or vice versa. If there is only one two-piece boiler, any reversal of flow does no harm as the water has no other place to flow than within itself. One cannot see the fluctuations.

The two-piece boiler, a fast circulator, is sensitive to any thing tending to vary the water level of one boiler from that of the other, to firing rate, load demand, or variance in nozzle size of any two oil burners.

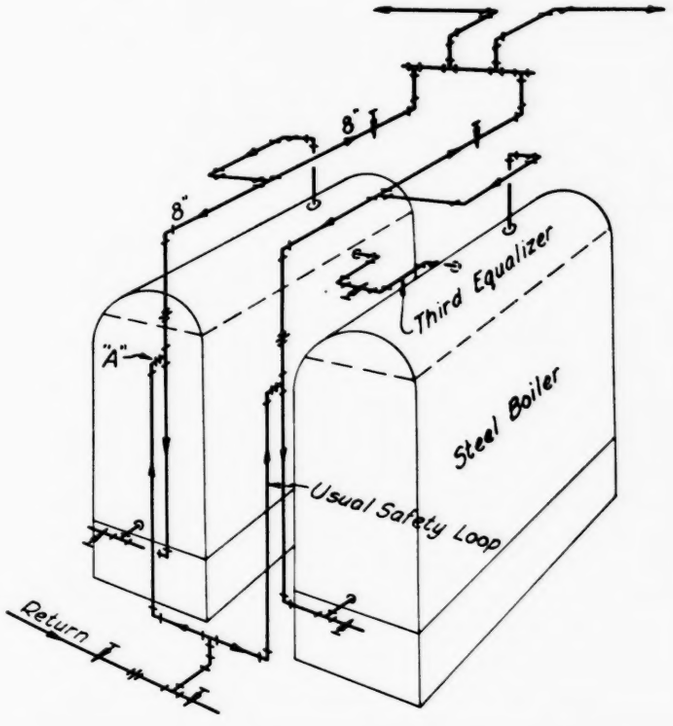

Fig. 1. Additional equalization for two-piece boilers.

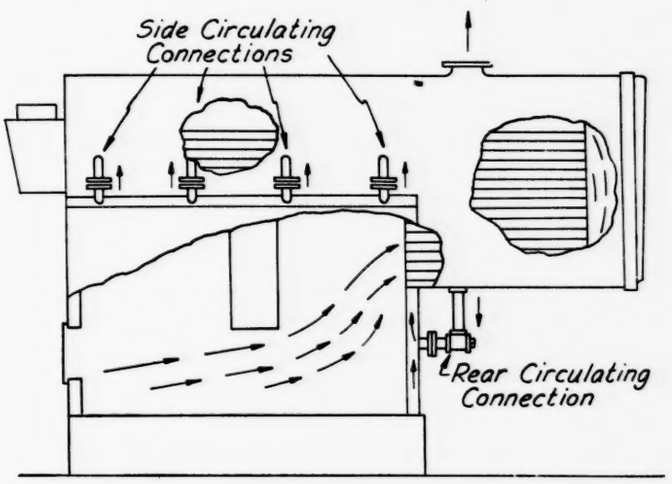

Fig. 2. Diagram showing arrangement of boiler of two-piece construction.

Difference in water levels occur in but a few moments. The same tendency to variance might be there in a one-piece or wrapper boiler, yet the variance would not show up rapidly. On one job with two hand-fired two-piece boilers, the fireman would open up the double firing doors of one boiler, then walk about 30 ft for his coal and, in a matter of 30 seconds, the inrushing cold air would chill the lower part of the boiler. This caused water to flow from one boiler to the other, but was finally corrected by the third means of equalization. This third means is simple, inexpensive, and a certain cure.

Theory of Operation

The theory back of the two-piece boiler is one of restricting passage of water between the firebox and the shell, since all of the upward surge must go through a number of small areas or side connections at high velocity, instead of through an unbroken opening following the entire perimeter of the firebox. For example, a two-piece boiler may have a total area between firebox and shell equal to but ten 2½-inch side connections for upward flow (5 per side) and one rear 2½-inch connection for downward or return flow. From the side connections, which vary from 2 to 3 ft on centers, the water fans out a bit to wipe the tube and shell heating surface free from steam-insulating

bubbles. Water velocity is also such that it reaches the top tubes and wipes them free of corrosive air bubbles which tend to collect there. Water close to the steaming point will quickly flash by its high velocity passing over the tubes, much the same as hot water will steam in a dishpan if stirred up by a spoon.

The ratio of total area of side connections to rear connection is 10 to 1, but this does not mean that the water is returned through the rear connection at a 10 times increase in velocity. Obviously, the side connections are delivering water for evaporation to meet the boiler load, and such water is not returned to the firebox. Nor is the velocity through the side connections the same for each. It is much faster in the connections nearest the boiler front. As a matter of fact, at times the water goes up through the front side connections while some of it returns down through one or more side connections near the rear of the boiler.

The two-piece boiler has its advantages. Obviously, two pieces can go through small openings and along narrow passages with greater ease than one large unit. Two pieces result in greater convenience in handling during shop fabrication, as well as economy in manufacture. However, these things are incidental. The original reason for two-piece boiler design is to meet boiler code requirements without providing an excess thickness of plate in the firebox. The boiler code calls for a boiler to have certain minimum thicknesses—¼ inch when the shell is less than 36 inches in diameter, and 5/16 inch and up according to greater shell diameters. The code also stipulates that ¼ inch is the minimum thickness of any boiler plate under pressure. It is then obvious that a firebox built separate from the shell need be constructed of plate only ¼ inch thick, regardless of how thick the plate in the shell must be. This makes quite a saving in steel plate.

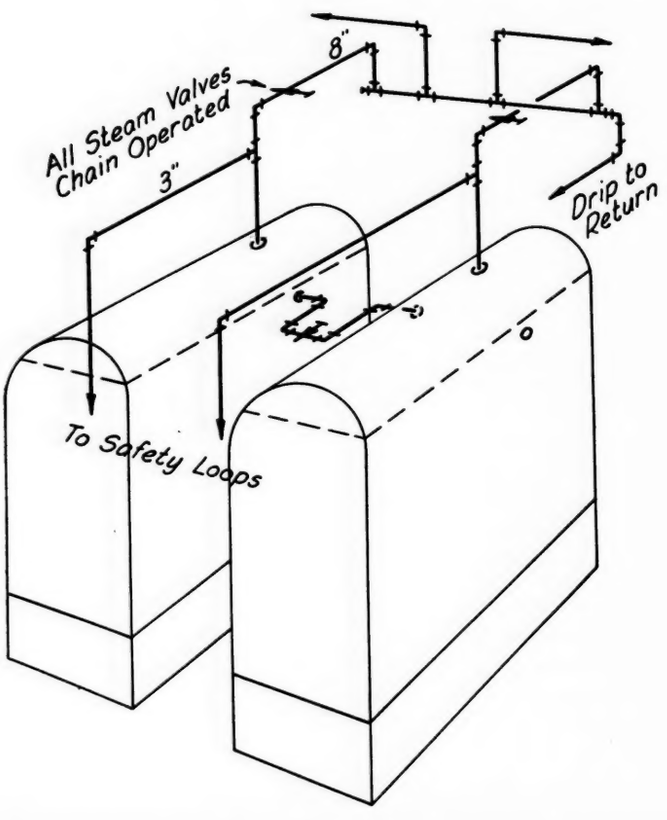

Fig. 3. Correction of the piping arrangement shown in Fig. 1.

While Fig. 1 and 3 illustrate the third means of equalization, Fig. 3 also shows how to alter the boiler piping connections to economize on piping and headroom and to provide better drainage and expansion for the piping. In Fig. 3 the pipe going to the rear need be only 3-inch instead of 8-inch. Drainage of condensate is not counter to the flow of steam and the steam riser from boiler outlet can go to the higher point to avoid lifting of water. Less piping is used and a means for expansion is freely provided.

With layouts shown in Fig. 1 and 3, the valves should be chain operated and the third equalizer sized 2-, 3-, or 4-inch for respective SBI boiler ratings of 3,000, 12,000 and 30,000 sq ft. The given equalizer sizes are minimum, but should be larger if the size of surface blow tappings permit. The top of the equalizer pipe should be set 1 or 2 inches below boiler water line. The gate valve in the equalizer can be turned upside down for convenience in operation. In this position it will not trap water when drained. As for the usual safety loops at the rear of boilers, these can be sized in the regular way. They should be arranged as shown for separate drainage, the house return likewise.

When a safety loop receives pump returns, it should be fitted with a close nipple at *A* as shown in Fig. 1, otherwise there should be a wye connection at this point. I have seen a few safety loops with the pipe at point *A* entering the boiler above its water line. The theory in this case is that any air from the return system will be liberated above the boiler tubes from whence it can travel out to the system for ultimate relief at some other point. The intent is to keep corrosive air bubbles away from the boiler tubes, but this is a theoretical contention, not a practical one, since the return water, being colder, reabsorbs air and does not liberate it.

Other Causes

Some other causes of difference in water line of two boilers, when oil fired, may be due to oil being delivered somewhat colder to one of the boilers, or to oil burner nozzles not calibrated for equal flow. Some installers try to synchronize the pressurestats to come on or go off at the same time. This is hard to achieve, especially as a system ages, and it is better to have but one burner on at a time. Since there is less stack loss with a smaller fire, there is more economy all around.

Another factor said to cause fluctuation of boiler water lines is that the boiler which steams first has a lot of light steam bubbles at its evaporating surface, hence to equalize the weight of water in one boiler with that of the other, the water line in the steaming boiler must be higher than in the other boiler. If there is a hot water coil in one boiler, it is quite apt to condense all steam as formed in this boiler while the other boiler steams and thereby contains lighter water. The boiler with the coil may be set for operation by aquastat control for 180F boiler water, while the other boiler may be controlled for operation at 5 lb steam pressure. Such a method will also cause inequality of any two boiler water lines.

Methods for Supporting Pipe

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Part 6 of a group of articles that describes and illustrates the various means for supporting pipe lines. This article covers spring hangers for piping and special hanger applications.

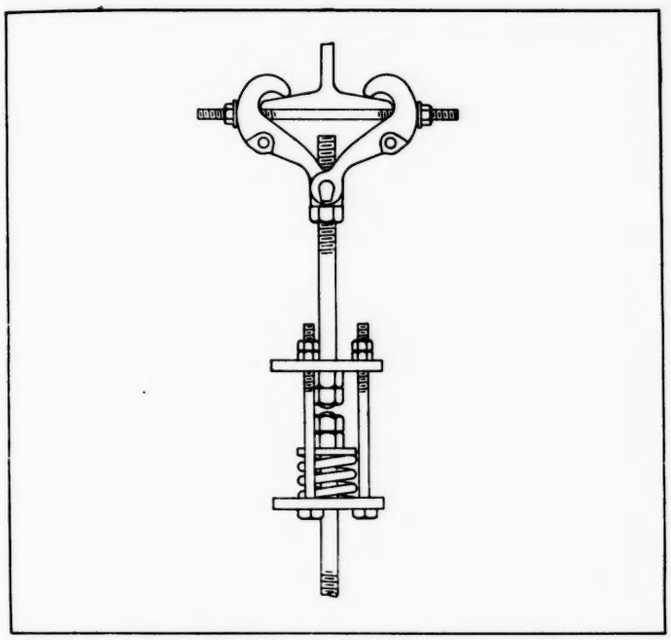


Fig. 89. Spring cushion pipe hanger.

Spring Hangers

Many forms of spring hangers are available. A spring cushion pipe hanger is shown in Fig. 89 and a spring cushion pipe roll anchor in Fig. 90. A single spring is used in Fig. 89 and two springs in Fig. 90.

The proper combination of springs to be used can only be determined after a careful analysis of the resulting stresses and anchoring forces, due to the combination of dead load and movements of the pipe due to expansion. For information on the method of making such an analysis, the reader is referred to Design for Flexibility by E. A. West and S. Smith of the Blaw-Knox Co., and to Hanger Load Calculations by Grinnell Co., Inc.

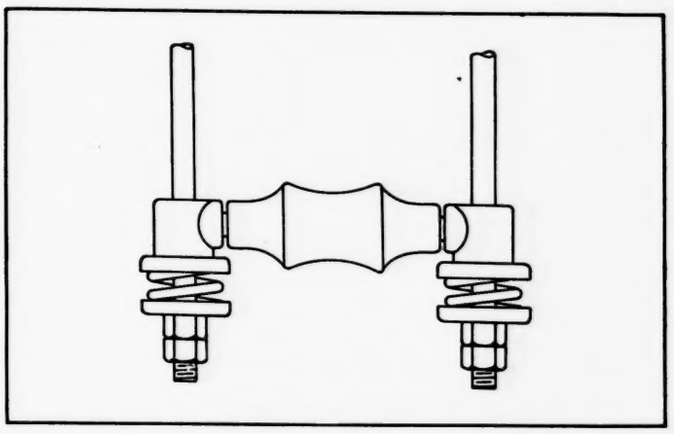


Fig. 90. Spring cushion pipe roll anchor.

In Fig. 91 is shown diagrammatically a section of a steam main with an expansion loop and two branch connections to equipment, and the necessary hangers, anchors and vibration eliminators installed in accordance with the results of such an analysis.

The Blaw-Knox functional hangers indicated on this diagram are shown in Fig. 92 and consists of a totally enclosed and guided spring with a pivotal connection within the hanger itself. This permits the portions of the hanger attached to a fixed support and to the pipe line, respectively, to pivot between extreme positions at considerable angle to each other (swivel action).

The hanger loading is adjusted by means of a turn-

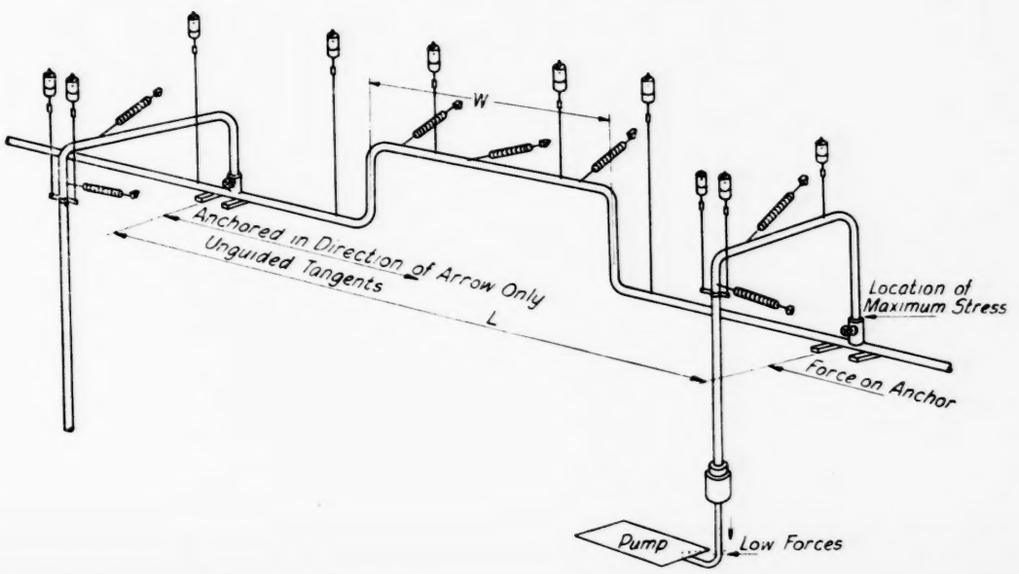


Fig. 91. Diagram of use of power piping functional hangers. When W = 0.5L most economical installation conditions exist. Free flexure of entire piping system is permitted in all planes.

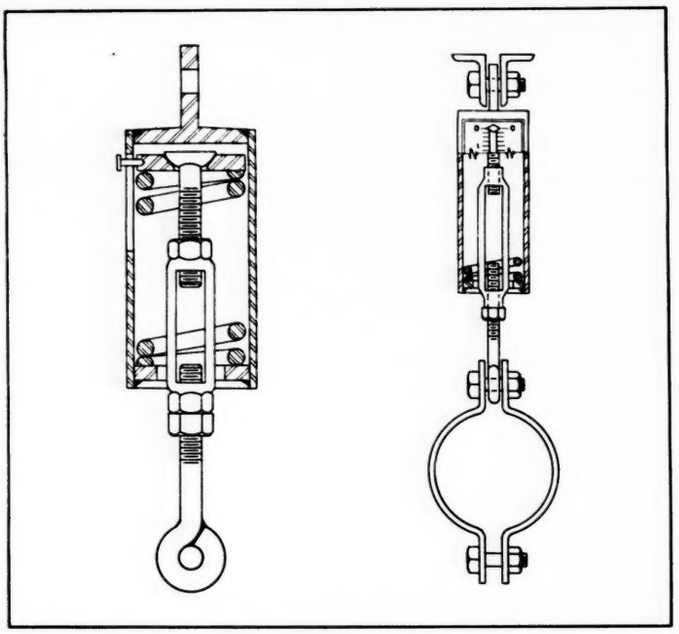


Fig. 92 (left). Details of hanger shown diagrammatically in Fig. 91. Fig. 93 (right). One application of hanger illustrated at left.

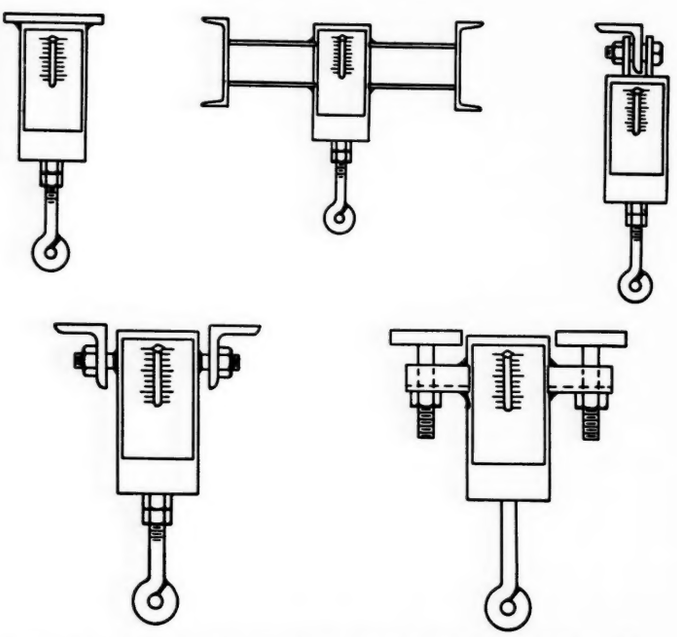


Fig. 94. Less usual arrangements of the hanger in Fig. 92.

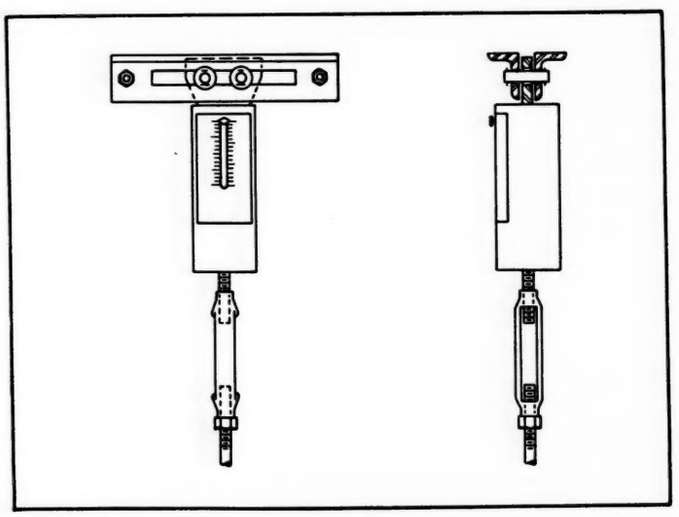


Fig. 95. Hanger that permits longitudinal and vertical movement.

buckle. A load scale, and load indicator provides means for determining the proper initial compression of the spring and for indicating the actual load and vertical movement at all times.

A typical installation of this type of hanger is shown in Fig. 93. Some less usual arrangements are shown in Fig. 94.

If it is desired to provide for longitudinal movement of the pipe, as well as vertical movement, rolls either as shown in Fig. 95 or 96 may be provided. If it is

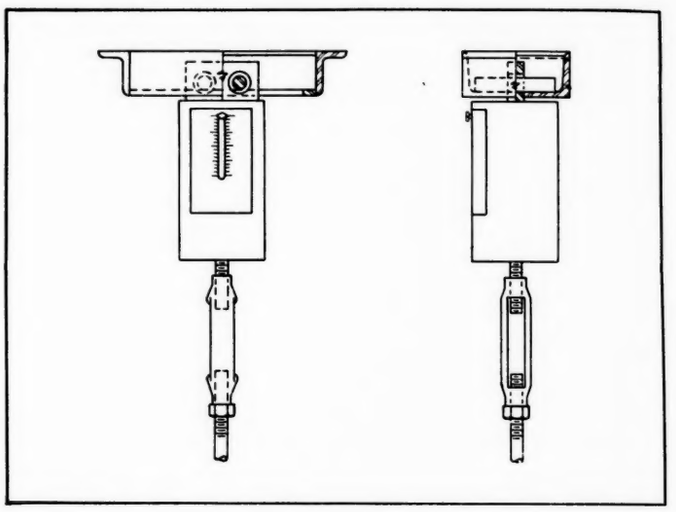
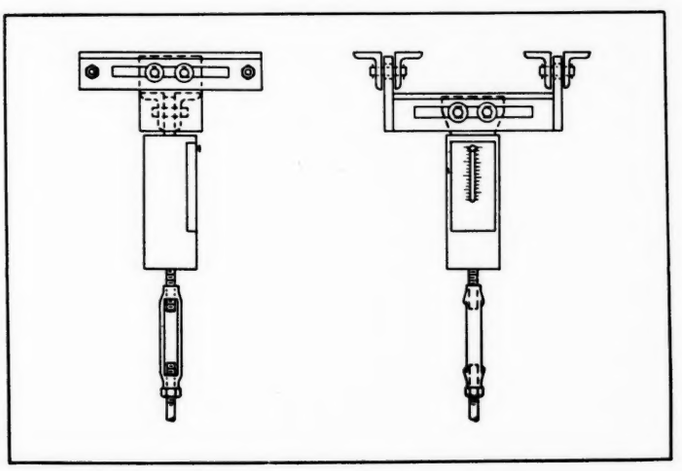
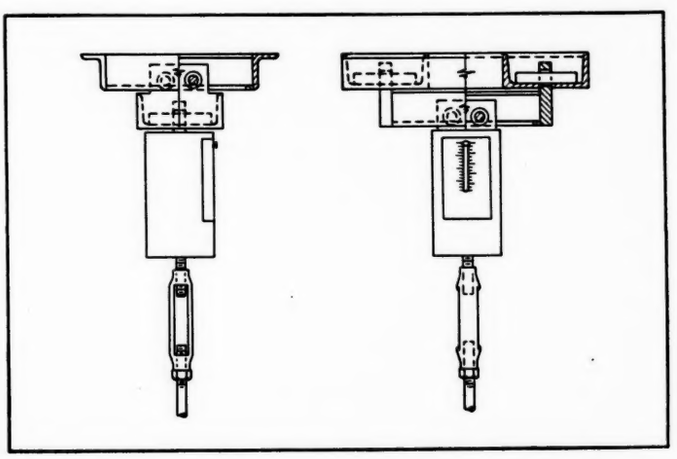


Fig. 96. Another type of hanger which will permit longitudinal as well as vertical movement of the pipe.





Where lateral movement is required, in addition to longitudinal and vertical movement, then designs shown in Fig. 97 (top) and Fig. 98 (bottom) may be used.

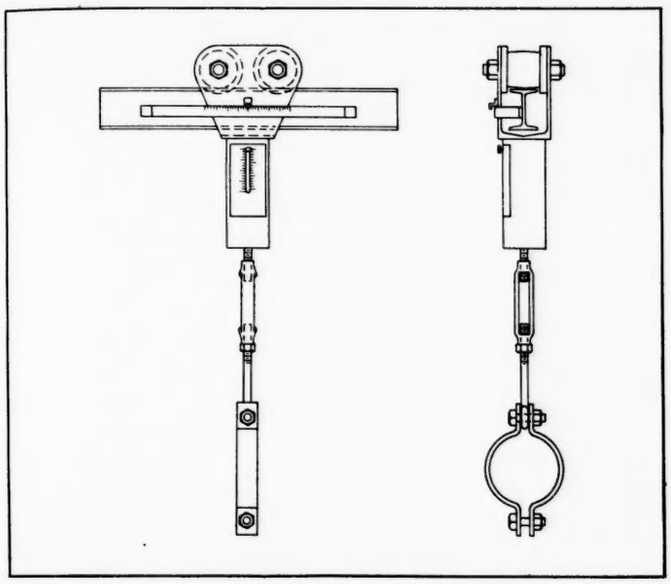
necessary to provide for lateral movement in addition to longitudinal and vertical movement, hangers of the multiple rolling action type, as shown in Fig. 97 and 98, may be provided. A pipe clamp is used for these hangers similar to that illustrated in Fig. 93.

Special type hangers with roller bearings lubricated through Alemite fittings, as shown in Fig. 99-102, may be provided to insure perfect freedom of horizontal movement. All of these types are fitted with horizontal and vertical travel scales, so that lines may be checked for change of cold spring, slipping of anchorage and "creep" which becomes more and more important as temperatures and pressures become more extreme.

Some special applications of multiple rolling units are shown and described as follows:

Fig. 103 shows an arrangement to meet extremely restricted headroom.

Fig. 104 shows a special method of mounting the



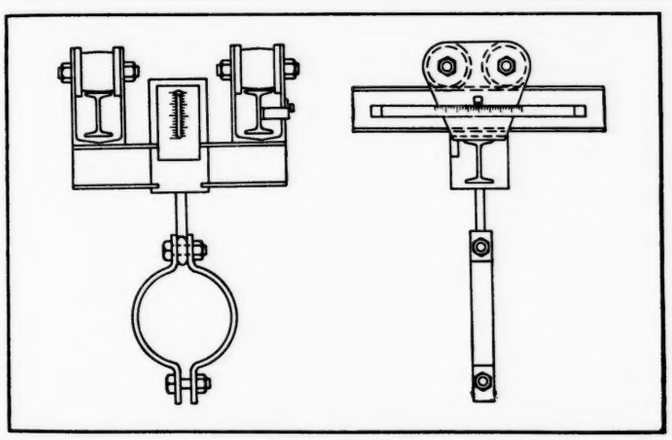


Fig. 99 (top) and Fig. 100 (above). Hangers to provide for horizontal movement in one line only. Fig. 100 is used where headroom is limited.

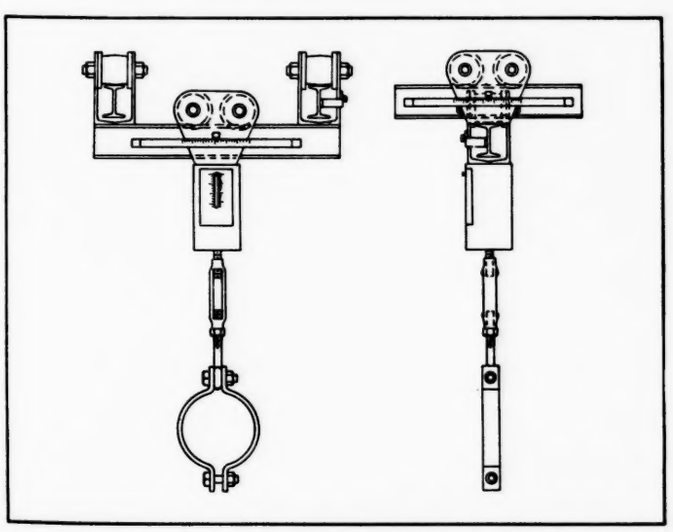


Fig. 101. Hanger that permits horizontal movement in two directions.

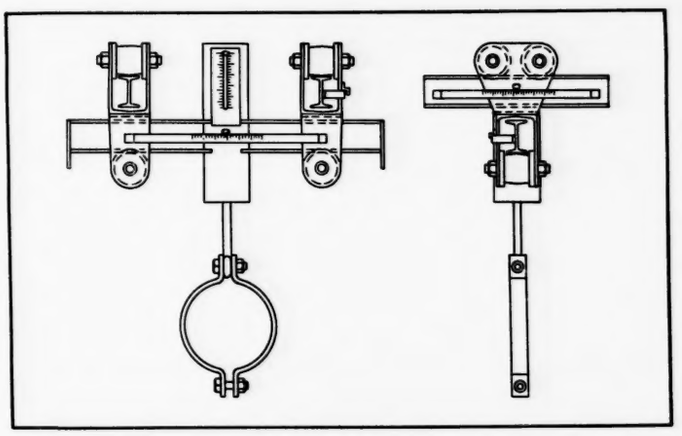


Fig. 102. Hanger, for use where headroom is limited, that permits horizontal movement in two directions.

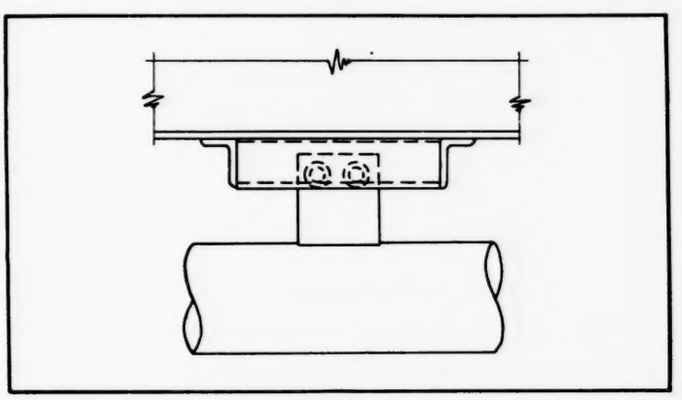


Fig. 103. Support used where headroom is limited.

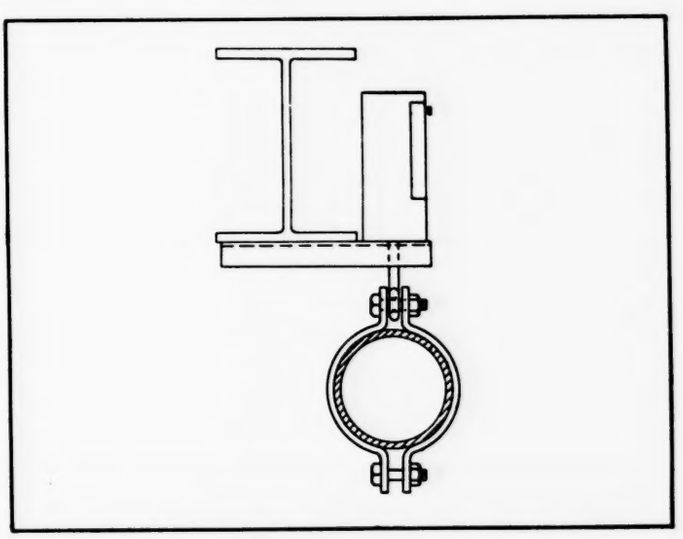


Fig. 104. Arrangement of spring hanger for use where there is little clearance.

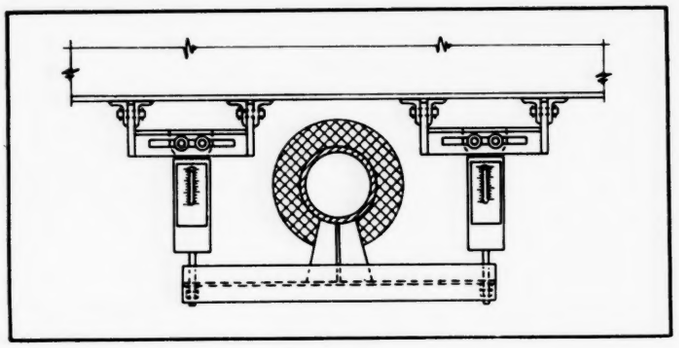


Fig. 105. Another use of spring hangers where there is a minimum amount of headroom.

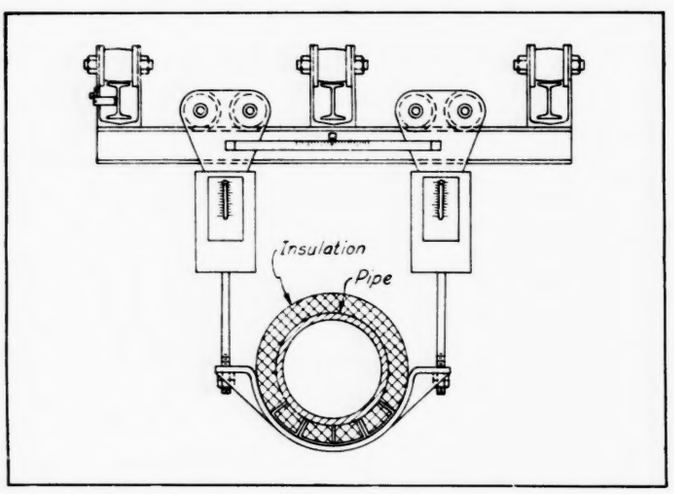


Fig. 106. Method of holding down seismic disturbances.

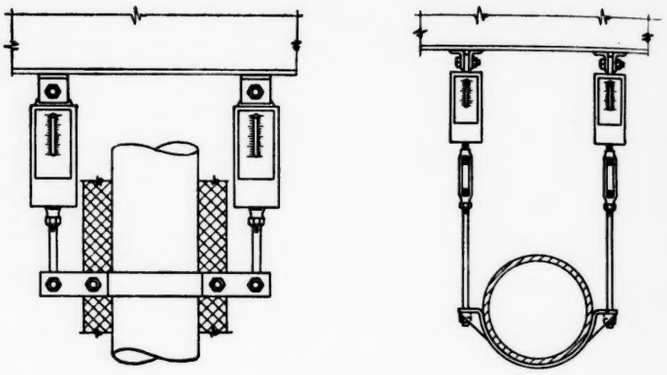


Fig. 107 (left). Use of spring hangers with a vertical pipe.
Fig. 108 (right). Special support for horizontal pipe.

spring hanger where there is limited clearance below the top of the pipe and a beam.

Fig. 105 was designed to meet extremely restricted headroom and at the same time to provide for lateral, horizontal and vertical movements.

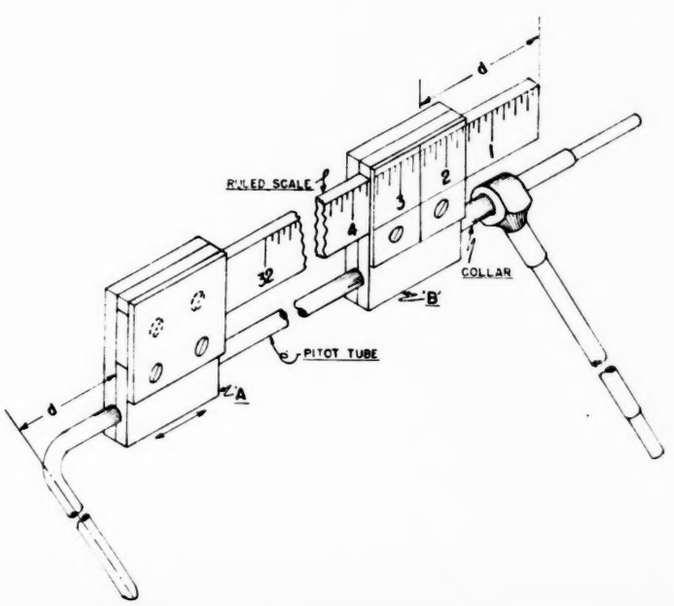
Fig. 106 shows multiple rolling units with further compounded rolling action to minimize the effects of seismic disturbances.

Fig. 107 shows a method of installing spring hangers to support a vertical pipe. The spring cases are sufficiently clear of the pipe covering to allow for pipe movement. Fig. 108 shows a method of supporting a horizontal pipe which offers minimum interference with other piping.

Scale Attachment Shows Position of Pitot Tube in Duct

Industrial hygiene engineers and others in the field of industrial ventilation use the Pitot tube for determining duct velocity and calculating the quantity of air flow handled by a ventilating system. W. H. Baumann, industrial hygiene engineer at Oak Ridge National Laboratory, has devised an inexpensive scale attachment that facilitates the use of the Pitot tube for field measurements by indicating the depth of penetration of the tube into a duct. A diagrammatic sketch of the scale attachment and Pitot tube is shown in Fig. 1. The entire scale assembly was made in the Oak Ridge carpenter shop. The material of construction was Plexiglass; however, other materials could be employed. The scale can be graduated as desired. The scales shown were made up on $\frac{1}{8}$-inch graduations similar to a conventional yard stick. For accuracy, the scale may be graduated on an engraving machine; however, our scales were marked by hand. Various ruled scales can be made up for different sizes of Pitot tubes.

The scale, mounted on the two end assemblies, *A* and *B*, is bolted to *A* and slides through *B* which rests against a $\frac{1}{2}$-inch brass collar. In operation, *A* moves along the Pitot tube as it is inserted into the ventilation duct. The distance *d* that the tube is inserted in the pipe is read off at the hairline on the cover of *B*.



Tested Practice in Steam and Hot Water Heating for Houses

A Resume of Research at the University of Illinois

Prepared by

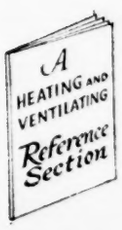
WARREN S. HARRIS
Special Research Associate Professor

and

ROBERT H. WEIGEL
Special Research Assistant

Part 2

Heat Transfer • Humidity • Operation
Control • Efficiency



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Last month, in Part 1 of this two-part condensation of almost half a century of research in steam and hot water heating at the University of Illinois, the authors covered heat emission and distribution from radiators, convectors, and radiant baseboards. In Part 2, concluded in this issue, heat transfer and humidity, operation and control, domestic hot water, and overall efficiency are discussed.

Part 2

Heat Transmission Through Glass

Surface temperatures for common glass subjected to approximately —2.3F outdoor temperature and a 10 mile an hour wind velocity, are shown in Fig. 19 for windows, both with and without storm sash (3). An air temperature of approximately 70F at the breathing line was maintained by a small-tube radiator located under the window.

The drop in temperature through the pane of glass from outside surface to inside surface when no storm sash was used was of the order of one degree, and the resulting temperature for the inside surface of the glass was 17.6F as compared to approximately 52F for the wall. Because of the low surface temperature in the case of unprotected windows, the radiation loss from room occupants to the windows is greater than to a corresponding area of exposed wall and, therefore, unprotected windows are more detrimental to comfort than the same area of cold wall surface.

Fig. 19 shows that when storm sash was placed on the window, the temperature of the inside surface of the glass rose to 41.3F. The presence of the storm sash, by raising the inside surface temperature, would therefore reduce the radiation loss from the occupants to the glass surface. Hence, an increase in the degree of comfort was noted when the storm sash was installed.

In addition to radiant effects, it may be noted from Fig. 19 that when the storm sash was not present, the temperature of the air within one inch of the inside surface of the window was only 54.0F as compared with 69.1F at the breathing level at the center of the room. In the case with the storm sash this layer of air was at a temperature of 60.8F. Thus, even though the presence of properly installed storm sash increased the surface temperatures of the inside surface of the glass, the need of placing the heating unit under the window so that upward currents of warm air can intercept these downward currents of cold air again becomes evident (see *Radiator Location*) if desirable air temperatures near the floor are to be maintained.

During the tests in the low temperature testing plant it was observed that the presence of curtains and

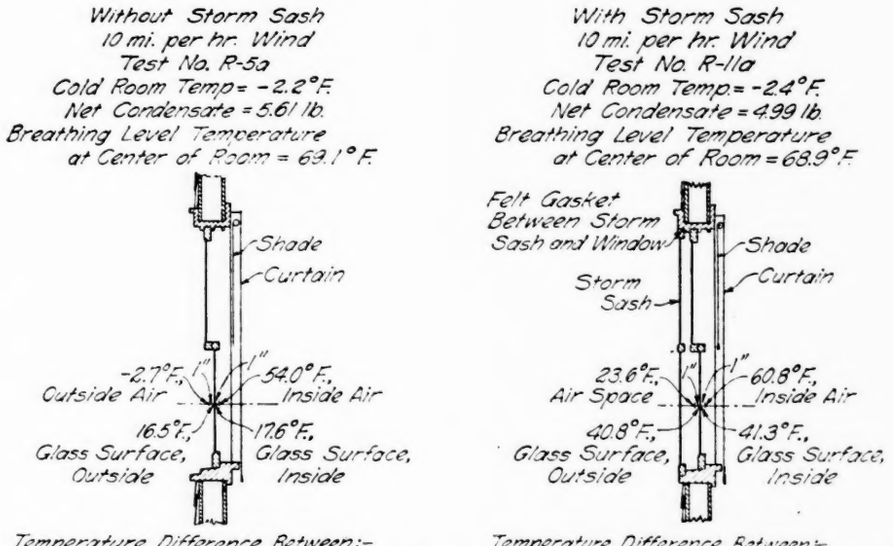

Fig. 19. Temperature gradients through windows.

shades did not materially influence the temperature of the inside surface of the glass. When no curtains or shades were present this temperature was about 19.5F and when the shades were tightly drawn it was about 12.4F as compared with the 17.6F, for partially drawn shades, as shown in Fig. 19. However, the curtains and shades did have a shielding effect which reduced the radiation from the room occupants to the cold glass, and a noticeable change in comfort was experienced even though no material changes in room temperature occurred.

Heat Transmission Through Radiator Recesses

Throughout the 1940-41 and 1941-42 heating seasons the temperatures of inside and outside surfaces of the sheathing directly back of the radiators in the dining room and the SW bedroom of the Research Home were observed (6). These records are of interest in that they show how the use of insulation in the radiator recesses may affect the heat loss from the house. The details of each of the wall sections used back of the radiator in these two rooms are given in Table 7. In Fig. 20 the observed temperature drops through the sheathing for each of the wall sections described in Table 7 are plotted against the indoor-outdoor temperature differences.

Since the conductance of the sheathing was the same in each case, the ratio of the temperature drops through the sheathing for any two sections is also representative of the ratio of the actual heat transmission through the sections. Therefore, for any given indoor-outdoor temperature difference, if the temperature drop through the sheathing for any given recess is divided by the temperature drop through the sheathing for the normal wall, a value which may be designated as the actual heat transmission ratio, "R", for that recess is obtained. This figure is also equivalent to the square feet of normal wall required to transmit the same quantity of heat per hour as is transmitted by one square foot of recess area. These actual heat transmission ratios have been calculated for each recess, and are shown as full-line curves in Fig. 21. The broken-line curves represent the ratio of the calculated overall heat transmission coefficient for each recess or wall section to that for the normal wall. The discrepancies between values read from the full-line curves and those read from the corresponding broken-line curves at given indoor-outdoor temperature differences result from the fact that the temperature back of the radiators was higher than that of the air in the room. The increase in the actual heat transmission ratios as the indoor-outdoor temperature difference increased was undoubtedly brought about by the fact that the higher aver-

TABLE 7—NORMAL WALL AND RECESS CONSTRUCTION

Construction Item	Normal Wall Section	Wall Section Back of Radiator				
		Recessed Radiator				Free-Standing Radiator
		Dining Room		Southwest Bedroom		Dining Room
		No. 1	No. 2	No. 3	No. 4	No. 5
Lath and Plaster	Yes	No	No	No	No	Yes
Vapor Barrier	Yes	No	No	No	No	No
3⅝-in. Mineral Wool Insulation	Yes	No	No	No	No	Yes
Reflective Insulation	No	No	Yes	No	Yes	No
⅞-in. Wood Sheathing	Yes	Yes	Yes	Yes	Yes	Yes
25/32-in. Rigid Insulation	No	No	No	Yes	Yes	No
15-lb Felt Building Paper	Yes	Yes	Yes	Yes	Yes	Yes
Air Space	½-in.	1-in.	1-in.	¼-in.	¼-in.	1-in.
Full Brick	Yes	Yes	Yes	Yes	Yes	Yes

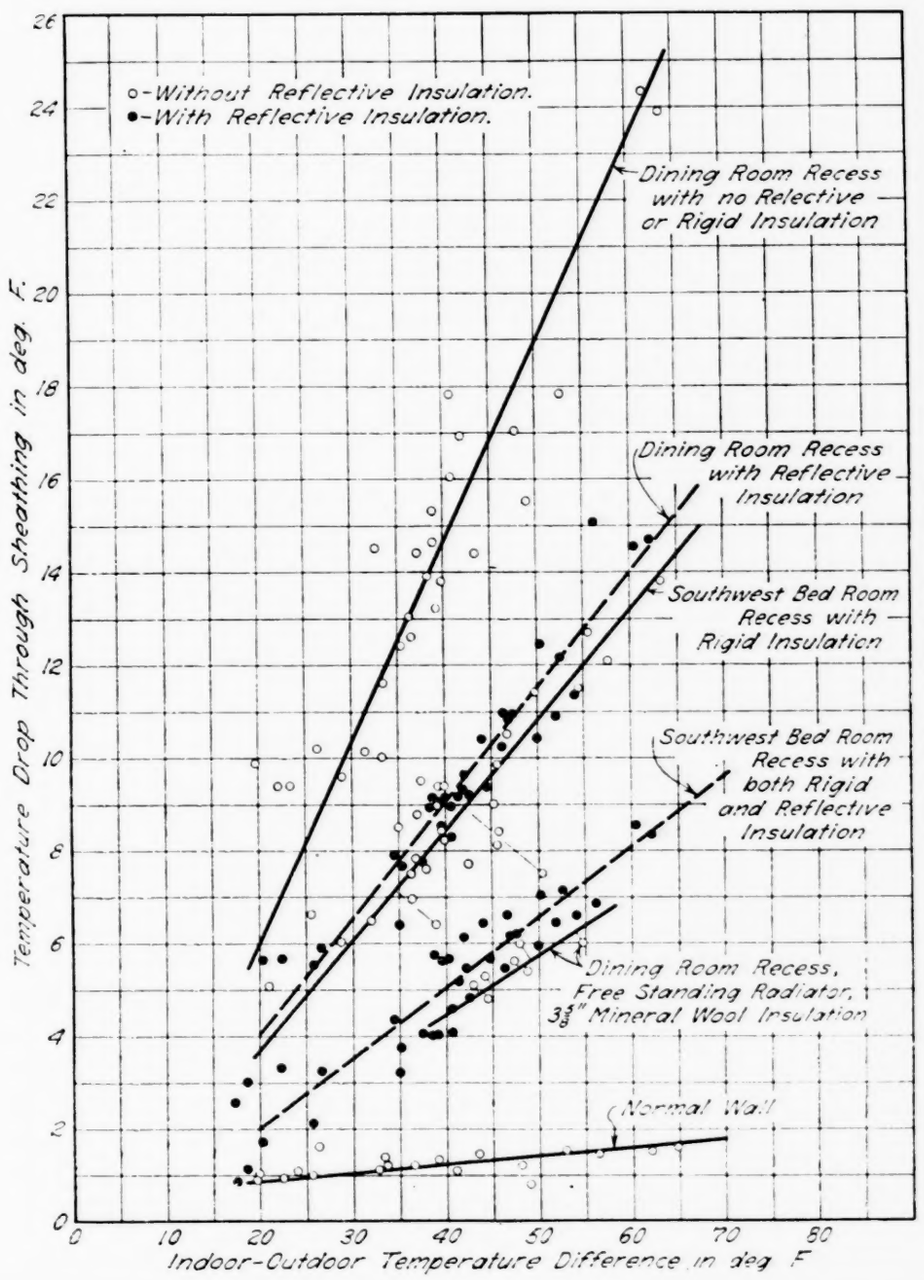

Fig. 20. Temperature drop through sheathing in radiator recess and normal wall.

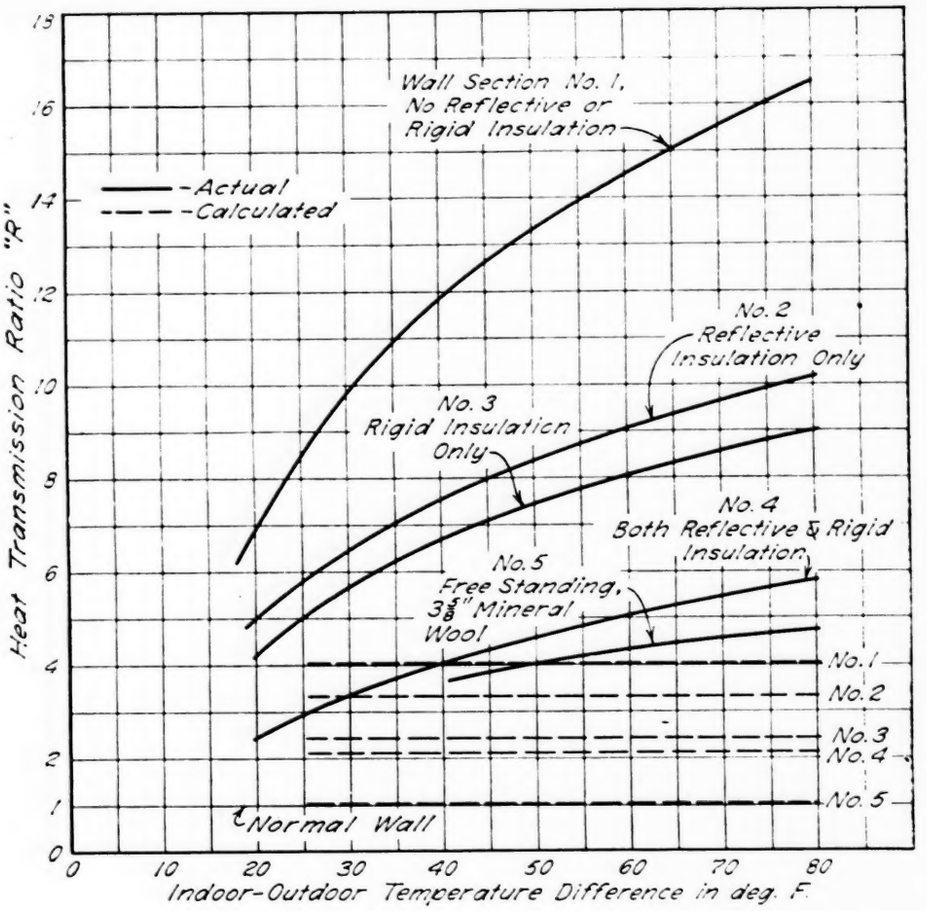

Fig. 21. Heat transmission rates for radiator recesses compared with that for normal wall.

age radiator temperatures required as the weather became colder resulted in greater differences between the temperatures back of the radiators and that in the room.

It is evident from Fig. 21 that considerably more heat was lost through an uninsulated recess than through a wall back of a free-standing radiator and that a large portion of this heat was saved by adequate insulation of the recess. It is also of interest to observe that reflective insulation on the surface of the recess was practically equivalent to 25/32 inch of rigid insulation, and, furthermore, that the heat lost through a recess insulated with a combination of the reflective and the rigid insulation was only slightly greater than that lost through the fully insulated brick veneer wall back of a free-standing radiator.

Certain values read from the curves in Fig. 21 are of particular significance from two standpoints. On one hand, since an indoor-outdoor temperature difference of 80F corresponds to the design temperature, values read at this temperature difference are directly related to the amount of radiation and the size of the boiler required. On the other hand, since an indoor-outdoor temperature difference of 34 degrees represents average winter weather in Urbana, Illinois, values read at this temperature difference can be correlated with the seasonal fuel consumption, and hence have a bearing on seasonal operating costs.

In the original calculations for the heat loss from the house, the 53.25 sq ft represented by the recesses were considered as normal wall and no additional allowance was made for individual peculiarities in the construction of the recesses themselves nor for the effect of the presence of the radiator. Columns 5 and 6 in Table 8 show the increases in total heat loss that would have occurred at indoor-outdoor temperature differences of 80 degrees and 34 degrees, respectively, due to the presence of the radiator and the type of wall construction, on the assumption that in each case all of the radiators used were installed in accordance with the designations in Column 1. The values in Columns 3 and 4 were read directly from the full-line curves in Fig. 21 at indoor-outdoor temperature differences of 80 degrees and 34 degrees. The values in Columns 5 and 6 were then computed from the following formula:

$$h = AUt_d (R - 1)$$

in which

h = increase in heat loss from house, Btu per hr

A = total area of recesses, (53.25 sq ft)

U = overall heat transmission coefficient for normal wall, (0.074 Btu per (hr) (sq ft) (F))

t_d = Indoor-outdoor temperature difference, deg F

R = actual heat transmission ratio for the given recess, at the temperature difference t_d

Column 7 shows that percentage increases of from 2.8 to 11.3 would have to be made in the amount of radiation installed to compensate for the increase in heat losses effected by the constructions indicated in Column 1. The percentages given in Column 8 at an indoor-outdoor temperature difference of 34 degrees are roughly correlated with the probable increases in seasonal fuel consumption, and show that due to the presence of the radiator and the type of wall construction, increases ranging from 1.68 to 7.08% might be expected in operating costs with the different types of recesses.

The ratio of the actual heat transmission through a recess or a wall back of a radiator to that through the normal wall, or the wall not influenced by the presence of a radiator, is generally assumed to be numerically equal to the ratio of the calculated heat transmission coefficient of the recess or wall back of the radiator to that of the normal wall. This, however, is true only if the actual conditions of heat flow and the actual details of the constructions exactly conform to those assumed in the calculations for the heat transmission coefficients. In making such calculations it is commonly assumed: 1—that steady flow actually exists; 2—that the air close to the wall has the same temperature as that in the main body of the room; and 3—that the inside surface of the wall is not subjected to radiation from any sources at a temperature higher than that of the air in the room. In the case of recesses, and walls back of radiators, subjected to cyclic action of both indoor and outdoor temperatures, none of these conditions is exactly satisfied. The heat flow is not steady, the temperature of the air close to the surface of the wall is not the same as that in the main body of the room, and the inside film coefficient is directly influenced by the close proximity of the hot radiator.

In order to derive a general relation between the actual heat transmission and the calculated values of the heat transmission coefficients, U for recesses and walls back of radiators, the actual heat transmission ratios have been plotted against the ratios of coefficients of the recesses, U_r, to the coefficients of the normal wall, U_w, as shown in Fig. 22. These curves are probably sufficiently general to permit the prediction of the approximate rate of heat transmission at the design indoor-outdoor temperature difference of 80 degrees for different types of recesses located in any given wall, and with mean water temperatures of approximately 170F in the radiators. It may be observed that the relation between the actual rate of heat transmission and the calculated value of U was influenced by the nature of the surface

directly exposed to the heat from the radiator. When this surface was covered with aluminum foil the actual rate of heat transmission with a given value of U was approximately 18% less than that obtained when aluminum foil was not used. Apparently the close proximity of the hot radiator had a less disturbing effect on the accepted values of the inside film coefficient in the case of a reflective surface than it did in the case of a non-reflective surface. Furthermore, it should be emphasized that the actual rates of heat transmission through walls back of radiators are much greater than the calculated rates obtained by using an indoor-outdoor temperature difference in connection with the commonly accepted values of the overall heat transmission coefficients.

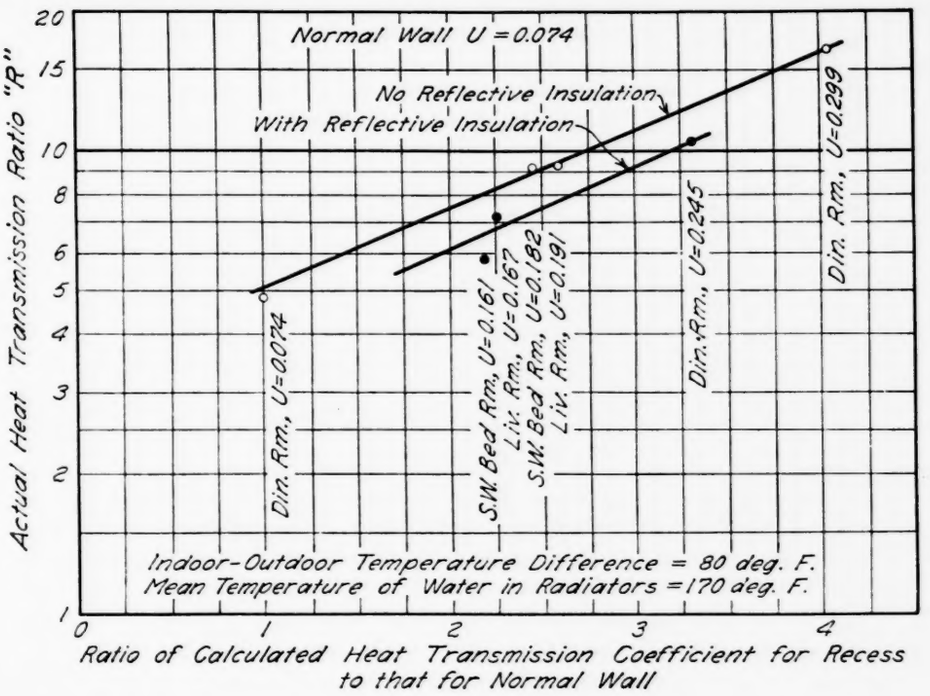
Actual Heat Transmission Ratio "R"
Normal Wall U=0.074
No Reflective Insulation
With Reflective Insulation
Din. Rm., U=0.074
S.W. Bed Rm., U=0.161
Liv. Rm., U=0.167
S.W. Bed Rm., U=0.182
Liv. Rm., U=0.191
Din. Rm., U=0.245
Din. Rm., U=0.299
Indoor-Outdoor Temperature Difference = 80 deg. F.
Mean Temperature of Water in Radiators = 170 deg. F.
Ratio of Calculated Heat Transmission Coefficient for Recess to that for Normal Wall

Fig. 22. Relation between actual heat transmission rates and calculated overall heat transmission coefficients.

Humidity in Houses in Winter

The problem of maintaining adequate indoor humidities for comfort in the winter cannot be separated from consideration of good building construction. In a tightly constructed house equipped with a vapor barrier on all exposed walls and ceiling, and indoor humidity adequate for comfort is often maintained without the assistance of a humidifier.

Fig. 23 shows the observed humidity records taken in the I=B=R Research Home and in a private home of similar construction. Both houses were built at the same time by the same contractor, have the same type of windows and vapor barrier, and have the same type of heating system with no provision for humidification. The houses are located in the same town within a mile of each other. The main difference in the operation of the two houses was that no family washing or cooking was done in the Research Home, while the private home had normal family usage.

Each test point represents the average humidity for a 24-hour period. While the test points for the private home are very scattered due to the fact that no control of operating conditions could be maintained, it is of interest to note that the observed humidity was definitely higher in the private home than in the Research Home. This was especially true in the colder weather. At an outdoor temperature of 0F the relative humidity in the private home was approximately 30%, whereas in the Research Home where no cooking, etc., was done it averaged only about 17%. During the winter 30% relative humidity is generally accepted as the lower limit for comfort. Thus, even though no humidifier was used in the private home in which these studies were made, the indoor air was sufficiently moist for comfort even when the outdoor temperature was as low as 0F.

Humidity observations were also made in a basementless house with no attempt made to increase the indoor relative humidity by means of a humidifier. The basementless house was occupied by a family, yet the relative humidities observed were in fairly close agreement with those obtained in the Research Home. Although both houses were insulated and provided with vapor barriers in the exposed walls and ceilings, and weatherstripping was applied to the doors and windows, the basementless house was not as tight in construction as the Re-

TABLE 8—HEAT LOSS INCREASES RESULTING FROM RECESSES

Method of Installing Radiator	Overall Heat Transmission Coefficient, Btu per (hr) (sq ft) (F)	Actual Heat Transmission Ratio[1]		Increase in Heat Loss From House, Btu per Hr		Increase in Heat Loss From House in Percent of Calculated Heat Loss[2]	
		80 deg F indoor-outdoor temp diff	34 deg F indoor-outdoor temp diff	80 deg F indoor-outdoor temp diff	34 deg F indoor-outdoor temp diff	80 deg F indoor-outdoor temp diff	34 deg F indoor-outdoor temp diff
1	2	3	4	5	6	7	8
Normal Wall	0.074	1	1	0.0	0.0	0.0	0.0
Free-Standing (No. 5)	0.074	4.8	3.3	1195	308	2.8	1.68
Reflective and 25/32-in. rigid insulation in recess (No. 4)	0.161	5.8	3.6	1510	349	3.5	1.90
25/32-in. rigid insulation in recess (No. 3)	0.182	9.0	6.1	2520	685	5.8	3.72
Reflective insulation in recess (No. 2)	0.245	10.2	6.9	2895	791	6.7	4.30
No insulation in recess (No. 1)	0.299	16.5	10.7	4880	1300	11.3	7.08

[1]Square feet of normal wall equivalent to construction shown in Col. 1; values taken from Fig. 21.

[2]Calculated heat loss of house at 80-degree indoor-outdoor temperature difference = 43,370 Btu per hr. Calculated heat loss of house at 34-degree indoor-outdoor temperature difference = 18,400 Btu per hr.

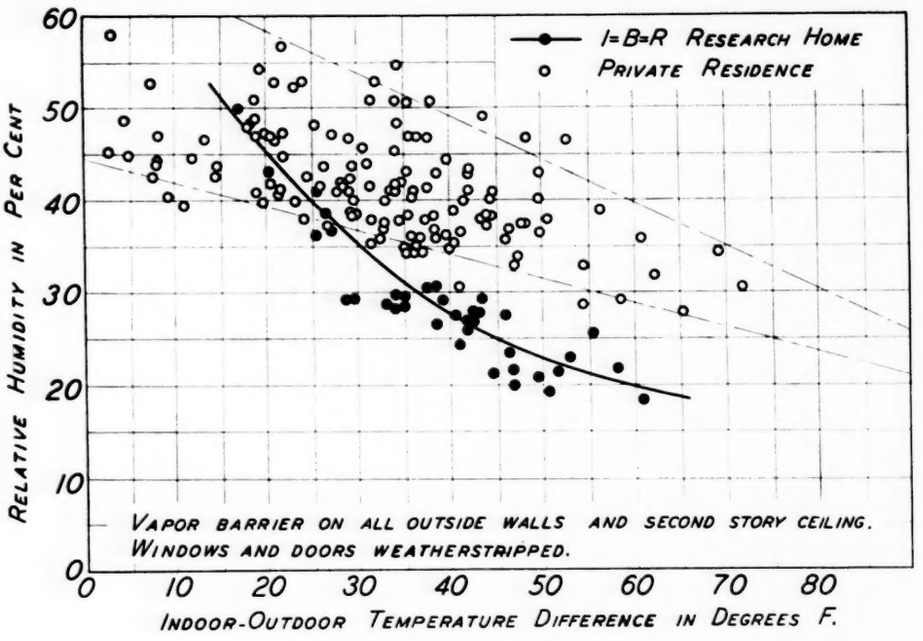

Fig. 23. Relative humidity in two houses with different occupancy; no special provision for humidification.

search Home. In the basementless house the outside door and the butted joints of the prefabricated wall panels were not tight, and possibilities existed for considerable air leakage and vapor transmission. Thus it is reasonable to assume that had the basementless house been of tighter construction, higher relative humidities would have been obtained. These observations are sufficient to demonstrate that the problem of maintaining adequate indoor humidities for comfort in winter cannot be separated from considerations of good building construction.

The maintaining of high humidity in well constructed homes sometimes results in excessive condensation on the windows. The lower curves in Fig. 24 show, for three types of windows, the indoor relative humidity at which condensation will begin to appear on the window panes as the outdoor temperature varies from —20F to 69F. With an outdoor temperature of 0F and an indoor temperature of 72F condensation on single glazed windows will occur if the humidity is above 14%. With tight fitting storm sash the indoor relative humidity may be as high as 40% before condensation on windows will occur. Thus, if humidity necessary for optimum comfort is to be maintained, it is highly desirable to install storm sash to prevent excessive condensation on the windows.

Methods of Operating Heating System

Various methods of operating a hot-water heating system have been tried in the I=B=R Research Home to see what effect these various methods would have on comfort and economy. Special tests included the reduction of thermostat setting at night; numerous ways of operating bedroom windows, doors, and radiation at night; the closing off of rooms; and the use of a fireplace to supplement the heating system.

Fuel Savings Resulting from Closing of Rooms

Any broad generalization that closing of rooms will result in material savings in fuel must be regarded with caution. In studies dealing with fuel utilization (5) the results are to a large extent dependent upon the specific arrangements of building, heating system, fuel-burning equipment, and control system. Hence, the results obtained from tests conducted in the Research Home do not apply exactly, and in the same order of magnitude, to all types of buildings, heating systems, fuels, and controls. Nevertheless, the results do offer some information in a field noticeably lacking in actual test data.

Two series of tests were made in the Research Home which differed only in the number of rooms being heated. In the first series all the rooms were heated with air temperatures 30 inches above the floor maintained at 72F both day and night. In the second series the radiators in the three second-story bedrooms were turned off, and the bedroom doors were kept closed.

For a wide range of outdoor temperatures, the closing of the three bedrooms on the second story actually resulted in an increase in fuel consumption over that obtained when the entire house was heated. At an outdoor temperature of 38F, corresponding to the average winter temperature in Urbana, Illinois, the increase amounted to approximately 2.5 lb of oil per day, or about 10%. For an outdoor temperature of 25F the fuel consumption for the two cases was about the same. At all outdoor temperatures below 25F some fuel savings were effected by operating with the bedrooms closed and unheated. Therefore, no saving in fuel consumption was accomplished for the heating season even though more than one-third of the living quarters were closed and not heated.

Even before any rooms were closed, the burner and boiler were largely oversized for the house. When the rooms were closed this condition was aggravated since it was not practical to reduce the heat input to the oil burner. Furthermore, the heat loss from the living room, in which the thermostat was located, was increased by an additional heat transfer through the ceiling to the unheated rooms. This resulted in an increase of the average water temperature supplied to the radiators and an increase in the frequency of circulator operation. These combined factors effected an increase in chimney losses. The increase in the chimney loss at an outdoor temperature of 38F, as determined by measurements of chimney gas temperature and quantity, amounted to 40,000 Btu per 24 hr, which is equivalent to about 2 lb of oil. This corresponded fairly well with the observed increase in fuel consumption at this same outdoor temperature.

Other tests were conducted in a second residence in Urbana, Illinois, in which the unheated rooms constituted one-fourth of the living quarters, but they were not adjacent to the room in which the thermostat was located. In addition, the heating plant contained provisions for maintaining the rate of heat input to the burner in relation to the heat loss from the house. Under these conditions a small saving in fuel consumption was effected.

Fuel Savings Resulting from Use of a Fireplace

The use of a fireplace in one room of a house under conditions in which the remainder of the house is maintained at a normal value is not conducive towards fuel conservation (5). During the period the fireplace is in operation, fuel consumption is only slightly less than if the fireplace is not in use. During the period when the fire in the fireplace is extinguished, however, the heated air in the house escapes through the open fireplace damper, with the result that a greater fuel consumption is required when the fireplace is used than when the fireplace is not used and the damper is kept closed.

When the room thermostat is located in the same room as the fireplace, however, the localized heating

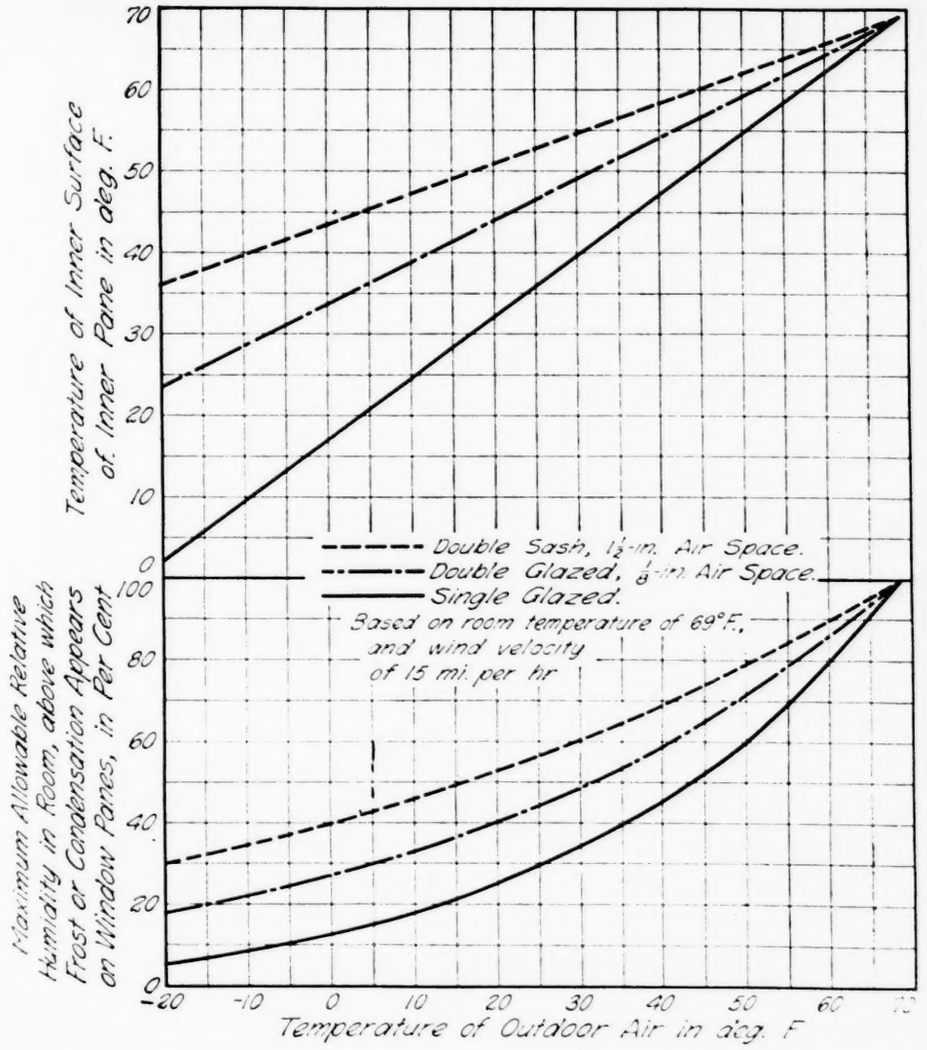
Temperature of Inner Surface of Inner Pane in deg. F.
Maximum Allowable Relative Humidity in Room, above which Frost or Condensation Appears on Window Panes, in Per Cent
Double Sash, ½-in. Air Space.
Double Glazed, ⅛-in. Air Space.
Single Glazed.
Based on room temperature of 69°F. and wind velocity of 15 mi. per hr.
Temperature of Outdoor Air in deg. F

Fig. 24. Glass surface temperatures and limiting values for relative humidity.

resulting from the fire helps to satisfy the demands of the thermostat and a substantial reduction in fuel consumption can be realized. Comfortable temperatures, in all probability, will not be maintained in any room of the house other than the room containing the fireplace. The reduction in fuel consumption is also proportionally much greater in mild weather than in cold weather.

In general, it may be concluded that in order to obtain the maximum fuel conservation with the main heating plant with the least disturbance to comfort conditions in the house, the following method of operation of the fireplace is desirable:

(1) Reduce temperatures, either manually or automatically, in all rooms in the house, other than the room containing the fireplace.

(2) Use the fireplace in mild weather, namely, in fall, early winter, and spring.

(3) Use the fireplace on relatively calm days.

(4) Close the fireplace damper when the fireplace is not in use, or place a tightly-fitting cover over the front of the fireplace after the fire has died down.

Operating House and Heating System at Night

The general operating conditions for tests of effects of reducing thermostat setting at night and of various ways of operating bedroom windows, doors, and radiation at night on comfort and economy (14) are given in Table 9; all tests were conducted in the Research Home. Except as noted in Table 9, bedroom doors were provided with felt weatherstrip at the bottom to prevent excessive leakage of air from the cold bedrooms to other sections of the house. In all tests in which the thermostat setting was reduced at night the changes in setting were made at 10:00 p.m. and 5:30 a.m. During tests with bedroom windows open at night one window in each bedroom was opened four inches at the bottom at 10:00 p.m. and closed at 7:00 a.m. Bedroom doors were always closed while bedroom windows were open, and in series D-46, series E-46, and series F-46 the bedroom radiators were turned off during the interval the bedroom windows were open.

Air Temperature Differentials with Reduced Thermostat Setting and Open Bedroom Windows at Night (14): Air temperatures during the day were not affected by turning off bedroom radiators, opening bedroom windows, or reducing the thermostat setting at night. Temperatures measured 3 inches above the floor and 3 inches below the ceiling of all second-story rooms during the period over which the house was maintained at 72F, were used as a basis of comparison. These temperatures were in close agreement during the mildest and coldest weather experienced, hence the various operating methods as described in Table 9 did not affect the floor to ceiling temperature gradient.

Morning Warm-Up Period with Reduced Thermostat Setting and Open Bedroom Windows at Night (14): To completely evaluate comfort under conditions of open bedroom windows at night, it is necessary also to consider the time required for the morning warm-up period. The morning warm-up time for the different series was assumed to be the total length of time the thermostat called for heat immediately after the change from night to daytime setting or, if the thermostat setting was not reduced, as in the case of series F-46, the total length of time the thermostat called for heat, starting with the closing of the bedroom windows.

For the series outlined in Table 9 it

TABLE 9—OPERATING CONDITIONS FOR TESTS WITH REDUCED THERMOSTAT SETTING AND OPEN BEDROOM WINDOWS AT NIGHT

Series	Thermostat Setting		Bedroom Radiators		Bedroom Doors		Bedroom[1] Windows		Radiation
	Day	Night	Day	Night	Day	Night	Day	Night	
B-46	72	66	on	on	open	open	closed	closed	RC
C-46	72	66	on	on	open	closed	closed	open	RC
D-46	72	66	on	off	open	closed	closed	open	RC
E-46	72	66	on	off	open	closed[2]	closed	open	RC
F-46	72	72	on	off	open	closed[2]	closed	open	RC

[1]Windows in rooms other than bedrooms closed during all tests.
[2]No weatherstrip.

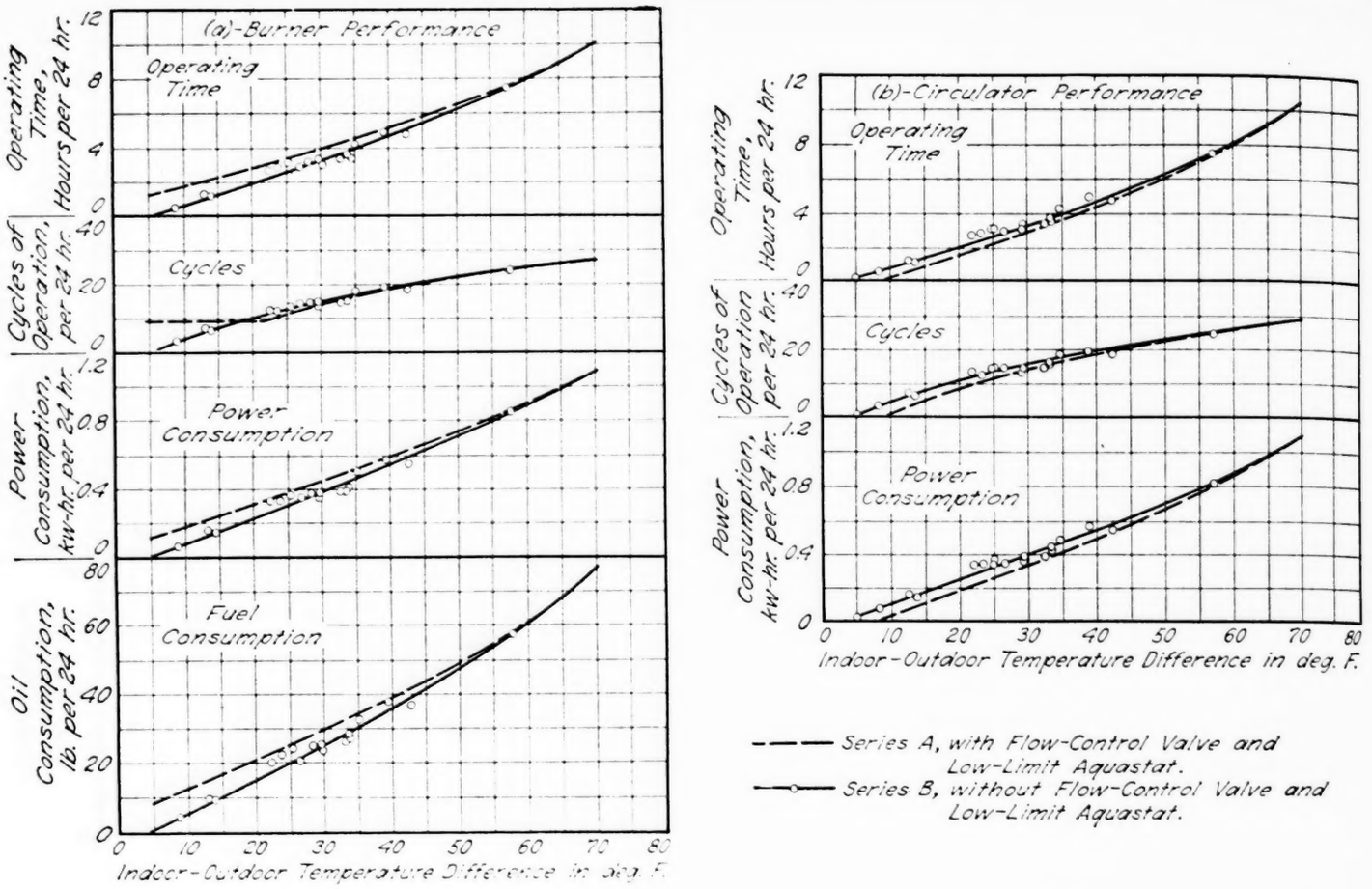
(a)-Burner Performance
Operating Time
Operating Time, Hours per 24 hr.
Cycles
Cycles of Operation, per 24 hr.
Power Consumption
Power Consumption, kw-hr. per 24 hr.
Fuel Consumption
Oil Consumption, lb. per 24 hr.
Indoor-Outdoor Temperature Difference in deg. F.
(b)-Circulator Performance
Operating Time
Cycles
Power Consumption
Indoor-Outdoor Temperature Difference in deg. F.
Series A, with Flow-Control Valve and Low-Limit Aquastat.
Series B, without Flow-Control Valve and Low-Limit Aquastat.

Fig. 25. Burner and circulator performance curves for two methods of control.

was found that for any outdoor temperature of 40F or higher the length of the morning warm-up period was about the same for series B-46, series C-46, series D-46, and series E-46. At outdoor temperatures below 40F, however, the curves representing the warm-up periods for these same series spread apart, with the longest warm-up period being required for series E-46, during which night setback was employed, the bedroom windows were open, and the bedroom radiators were turned off at night. On the other hand, when bedroom windows were open at night but bedroom radiators were not turned off, as in series C-46, the warm-up periods were shorter than for series E-46. The reduction in time increased with decreasing outdoor temperatures.

At all outdoor temperatures the warm-up time for series F-46 was considerably shorter than the warm-up time for any of the other methods of operation. Series F-46 did not call for a reduction in the thermostat setting at night.

While the warm-up times for those series employing night setback appear to be quite long, actually the radiant effect resulting from all radiation operating at maximum temperature, combined with the slight overrun in ceiling temperature, resulted in the occupants experiencing the feeling that the house temperature had reached the normal daytime level as much as two hours before the thermostat ceased to call for heat. Of course, the warm-up time for all series would have been reduced had excess radiator capacity been provided in each room.

Daily Fuel Consumption when Operating with Reduced Thermostat Setting and Open Bedroom Windows at Night (14): Reducing the thermostat setting at night but leaving all windows closed resulted in a daily fuel savings of approximately 5% as compared to operating with no reduction in thermostat setting. When bedroom windows were opened and the thermostat setting was reduced at night the daily fuel consumption was the same as that obtained when the windows were left closed and the entire house was maintained at 72F at all times. Furthermore, it seemed to make no difference as far as fuel consumption was concerned whether the bedroom radiators were turned off at night or not. Whatever was gained during the night by turning off the bedroom radiators was lost during the longer warm-up period in the morning.

Effect of Weatherstrip at Bottom of Bedroom Doors on Room Air Temperature and Daily Fuel Consumption when Operating with Open Bedroom Windows (14): As a rule, bedroom doors are not weatherstripped and they are normally fitted with about ¾-inch clearance between the bottom of the door and the floor. Series E-46 was undertaken to determine how leakage of air through this crack might affect room air temperatures and fuel consumption. This series differed from D-46 only in that in the former the weatherstrip was removed from the bedroom doors.

Adding weatherstrip to the bottom of the bedroom doors did not materially affect the second-story air temperature differentials or the length of the morning warm-up period. Nevertheless the daily fuel consumption for series E-46 was definitely higher than in series D-46 at all indoor-outdoor temperature differences over 20 degrees. This increase in daily fuel consumption was caused by cold air from the bedroom flowing through the cracks under the bedroom doors and on down the stairway into the living room, thus cooling the living room more rapidly at night and thereby increasing the heat requirements of the house throughout the night time period. From these tests it appears that if bedroom windows are to be opened at night, not only should the bedroom doors be closed during the time the windows are open, but, in the interests of economy, the bedroom doors should be weatherstripped or other suitable provision made to prevent the leakage of cold air from the bedrooms into the other rooms of the house.

Controls for Hot Water Systems

The principal objective of a series of tests of control systems for one-pipe, forced-circulation, hot water systems (6) was to compare the operating characteristics of an oil-fired, one-pipe, forced-circulation, hot-water heating system operated: (1) with a control system which would permit the boiler to be used to heat domestic hot water as well as to supply heat to the rooms; and (2) with a control system such that the boiler could supply heat to the rooms only.

The two series of tests mentioned above were conducted in the Research Home. For each series the amount of installed radiation was the same and the thermostat was set to maintain 72F at the 30-inch level both day and night. When the thermostat called for heat both burner and circulator were started and continued to operate until the thermostat was satisfied, unless the period of operation was of sufficient duration to heat the boiler water to a temperature of 200F, corresponding to the setting of the high-limit control. If this occurred, the burner would shut off by action of the high-limit control but the circulator would continue operating.

In series A, where the control system provided for the heating of domestic hot water as well as supplying heat to the house, a low-limit control prevented the temperature of the water in the boiler from dropping below 165F. A flow-control valve prevented any circulation of water between the boiler and radiators except when the circulator was in operation.

In series B, where no provision was made for heating domestic hot water, the low-limit control was eliminated and the flow-control valve was locked open so that it could not prevent the circulation of water. Under these conditions the radiators could continue to cool all of the water in the system as long as sufficient temperature difference persisted to effect any circulation by gravity, and the temperature of the water in the boiler could vary between that of the room and 200F. The operation of the burner and circulator was controlled by the room thermostat alone.

Burner and Circulator Performance (6): A comparison of the burner performance curves for series A and series B, Fig. 25, indicates that in mild weather, operation on series A resulted in increased burner operating time, burner power, and fuel consumption. These increases were accompanied by higher flue gas temperatures and larger heat losses from the chimney gases. As the indoor-outdoor temperature difference became larger, the average boiler water temperature for series B automatically approached that for series A as a result of the more frequent burner operation required to maintain the normal room temperature of 72F. At approximately 70 degrees indoor-outdoor temperature difference, the curve for series A becomes tangent to that for series B, and at all temperature differences above this the operating characteristics for both series were the same. This indicated that for indoor-outdoor temperature differences above 70 degrees the amount of burner operation required in series A to maintain normal room temperature was sufficient to maintain a minimum water temperature of approximately 165F in the boiler at the location of the low-limit control. In milder weather the room thermostat did not operate the burner over a long enough period to maintain the required minimum water temperature in the boiler, and, as a result, in series A the low-limit control continued the operation of the burner for a short time at the end of each cycle after the room temperature became normal and the circulator ceased to operate. The excess in the amount of fuel burned, power consumed, and time of burner operation for series A over that for series B is the amount of each required to maintain sufficient temperature in the boiler to make it possible to use the latter as a means of heating water for domestic purposes. Since the internal water heater was not connected to the hot-water storage tank during these tests, the results do not include any data on the stand-by loss from the storage tank, or on the requirements for the actual heating of water for domestic uses.

The increase in fuel consumption for series A over series B is directly related to the average temperature of the water in the boiler and would be affected by any changes in the setting of the low-limit control. Furthermore, the excess amount of fuel required to operate the boiler so that it can be used to heat water for domestic service may vary with the size of plant and the total load on the boiler.

No significant difference existed in the number of burner operating cycles required for series A and B except in very mild weather, in which case a minimum of 9 cycles per day was required in series A in order to maintain a minimum boiler water temperature of approximately 165F at the location of the low-limit control.

At any given indoor-outdoor temperature difference below 70 degrees the circulator operating time, power, and number of cycles were slightly less in series A than in series B. This condition resulted from the fact that higher average radiator temperature, and consequently higher rates of heat input to the rooms, were obtained during the on-periods of the circulator in series A than in series B.

Water Temperatures in Boiler and Radiators (6): Fig. 26 and 27 show the temperature variations of the water in the boiler and the radiators for a complete operating cycle representative of series A and series B. Water circulation was at the rate of about 13 gpm during the time of circulator operation. In series B when the circulator was not running some gravity circulation of water occurred, but at a rate too low to measure. In series A the flow-control valve prevented any circulation of water taking place between the boiler and the radiators during the off-periods of the circulator.

As indicated by the curves for series A (Fig. 26) the water in the boiler was still hot at the end of the off-

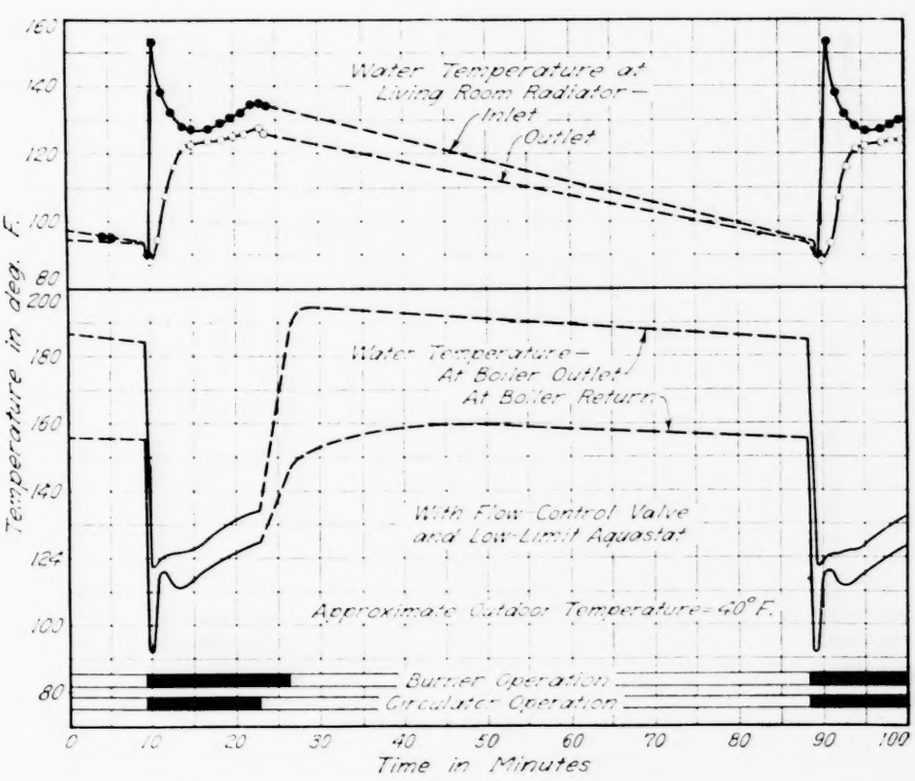

Fig. 26. Water temperatures, operating with flow-control valve and low-limit control.

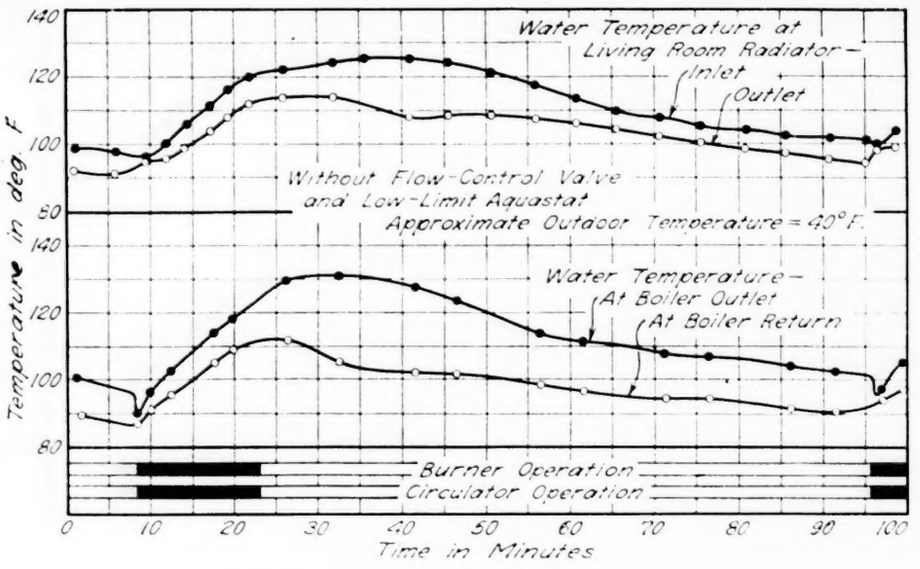

Fig. 27. Water temperatures, operating without flow-control valve and low-limit control.

period, while that remaining in the piping and radiators had cooled down to approximately 90F or 95F, depending on the length of the off-period. When the room thermostat closed the circuit, the circulator and burner were started simultaneously. Within the time of one minute the hot water which was stored in the boiler was distributed to the radiators in the house, quickly supplying heat to prevent further cooling of the rooms. At the same time, the cooler water contained in the piping and radiators during the off-period was returned to the boiler to be reheated. After the first surge of heat from the water stored in the boiler had been expended, the water entering the radiators dropped in temperature, owing to the fact that the burner, during the interim, could not heat the returned water to a temperature as high as that of the water originally stored in the boiler at the start of the on-period. The period represented by decreasing temperature of the water entering the radiators continued until all of the water contained in the piping and radiators during the off-period had passed through the boiler. After the period of dropping temperature, and extending throughout the rest of the on-period, there was a gradual increase in the temperature of both the water in the boiler and that supplied to the radiators, since the burner supplied heat to the water in the boiler more rapidly than it was removed from that in the piping and radiators.

In 13½ minutes the room thermostat stopped the circulator. At this time the average water temperature in the boiler was only 130F, and the low-limit control continued to operate the burner for an additional 3½ minutes, heating the boiler water to approximately 175F, or 10 degrees above the control setting. During the off-period of the circulator, with the flow-control valve in the system, the water in the radiators and uninsulated piping cooled about 40 degrees while that in the insulated boiler dropped approximately 10 degrees.

As shown in Fig. 27, the variations in water-temperature when operating on series B, without the flow-control valve and low-limit control, were somewhat different. In this case the boiler and radiators cooled at about the same rate during the off-period, since there was nothing to prevent water circulation by gravity action. When the rooms required heat, the burner and circulator were started simultaneously by the thermostat, and a gradual increase in water temperatures followed. After 14½ minutes of operation the room thermostat stopped both the circulator and the burner, even though the temperature of the water in the boiler averaged only about 118F. As a result of heat absorbed, largely from the residual heat stored in the refractory fire box, the temperature of the water leaving the boiler continued to rise from 123F, at the time the burner shut off, to 131F, some 9 or 10 minutes later.

Air Temperature Variations Within and Between Rooms (6): Although there was some possibility that the elimination of the flow-control valve might cause overruns in the air temperatures within the house, the maximum variation at any level during a cycle was only 0.9 degree, and this occurred near the ceiling. At the 30-inch level this variation amounted to only 0.5 degree. These values are well within the limits of control to be expected from the average room thermostat available today.

Two reasons may be assigned to explain why overruns did not occur when operating under the conditions existing in series B: (1) For given outdoor temperatures above 0F the average water temperatures in the radiators were lower in series B than they were in series A, and, therefore, the rate of heat output was more nearly equal to the rate of heat loss from the house in series B than it was in series A. For given outdoor temperatures below 0F the water temperatures and rates of heat output were approximately the same for both series. (2) The boiler used in these tests had a water capacity of 13.5 gallons, while the radiators and piping had a capacity of approximately 38 gallons. With either method of operation, the heat contained in the 38 gallons of water in the piping and the radiators at the time the room thermostat stopped the circulator, was released to the house during the off-periods. Thus, the only additional heat released during the off-periods in series B, over that released in series A, was the heat contained in the 13.5 gallons of water in the boiler and that which the water could absorb from the heated boiler and setting.

Temperatures measured 3 inches, 30 inches, and 60 inches above the floor and 3 inches below the ceiling in all of the rooms of the Research Home were in close agreement for both methods of control. In the coldest weather experienced, a difference of only approximately 4.5 degrees was observed between the average temperature of the air 3 inches above the floor and that 3 inches below the ceiling.

Heat Utilization and Overall House Efficiency

All the heat in the fuel the consumer buys for house heating finds its way eventually to the outdoors. Part of it reaches the outdoors directly as flue gas loss; the rest escapes through walls, windows, floors, and ceilings which are exposed on one side to the outdoors or to unheated spaces. Since the function of a heating system is to add sufficient heat to the house to offset the heat losses through walls, windows, floors, and ceilings, only that portion of the heat escaping directly from the house in the flue gases is a real loss. The remainder is ultimately utilized in offsetting the house heat losses (6, 11).

It is a popular conception that, for domestic heating systems, boiler or furnace efficiency is synonymous with overall heating efficiency. The figures usually quoted are of the order of 55 to 65% for hand-fired systems using bituminous coal, 65 to 75% for stoker-fired systems and systems converted to oil or gas, and 75 to 80% for systems using boilers or furnaces designed especially for burning oil or gas. An accurate analysis of the total heat utilization of central heating sys-

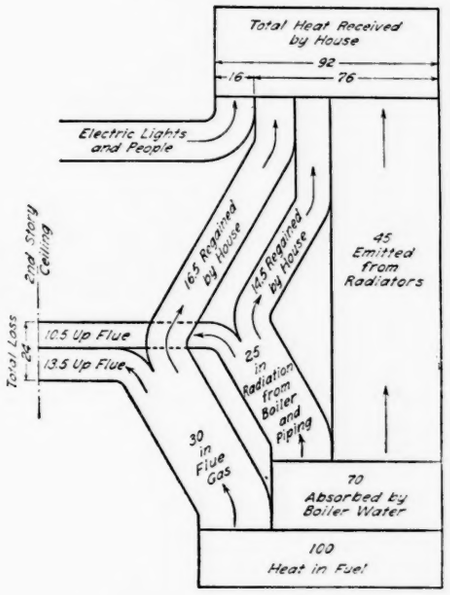

Fig. 28. Heat flow diagram obtained from tests on a forced-circulation, hot-water heating system in the I=B=R Research Home.

tems in actual use in the Research Home and Warm Air Research Residence No. 1 reveals that the boiler or furnace efficiency is not necessarily an index of overall house efficiency, but rather that any well-designed heating system which is thermostatically controlled will operate at an overall efficiency of approximately 80% regardless of whether the system is hand-fired, stoker-fired, or fired with oil or gas. A large part of the so-called heat "losses" from heating plants are actually utilized in helping to warm the house. The high overall efficiency quoted above is for a heating plant operating in a basement and might have to be modified slightly for a system operating in the utility room of a basementless house, or if the heating plant utilized a chimney on an outside wall rather than an inside chimney, since it is possible that under either of these conditions a smaller percentage of the heat losses from the boiler and distributing system would be utilized in warming the house.

Heat Flow

A heat flow diagram obtained by tests on a forced-circulation, hot-water heating system installed in the Research Home is shown in Fig. 28. From the daily observations of flue gas temperature and per cent CO_2 at the smoke collar of the boiler it was determined that about 70 heat units out of every 100 released by the combustion of the fuel were absorbed by the water in the boiler and that about 30 units were discharged from the boiler as waste heat in the flue gases. Thus the boiler efficiency during the tests was approximately 70%. Of the 70 heat units absorbed by the boiler, about 45 were transmitted to the radiators; the other 25 were "lost" by radiation and convection from the boiler and piping. In other words, out of every 100 heat units supplied to the boiler by the fuel burned, only 45 were delivered to the rooms by the radiators. However, it can be shown that a large proportion of the 55 heat units supposedly lost were actually utilized in helping to warm the house.

In order to determine the total heat input to the house, daily records were made of the pounds of oil used by the burner, the kilowatt hours of electricity used in lights and power, the cubic feet of gas burned in an auxiliary water heater, and the man-hours of occupancy. The heat equivalents used for all of these sources of energy were as follows:

Oil—19,550 Btu per lb
Electricity—3,415 Btu per kw-hr
Gas—1,000 Btu per cu ft
People—500 Btu per man-hr of occupancy.

It was assumed that all the heat represented by the electricity used, and that supplied by the occupants, was available as useful heat. The only appliance using gas was an auxiliary water heater. This heater was connected to the outside chimney, and it was estimated that 25% of the total heat equivalent of the gas used was lost in the flue gases. Hence, only 75% of the heat equivalent of the gas burned was considered available as useful heat.

By determining the average temperature and rate of flow of the flue gases at the level of the upper boundary of the heated space for both the on- and off-periods of the burner, the heat actually lost at the top of the chimney was determined. These losses during the on-period consisted of the heat lost in the dry flue gases, water vapor formed by the combustion of the hydrogen in the fuel, and any water vapor carried in by the air supplied. During the off-periods the loss consisted only of the heat lost in the dry flue gases and in water vapor carried in by air. The difference between the heat lost at the top of the chimney and the heat in the flue gases at the smoke collar represented heat that was transferred to the air in the house through the chimney and flue pipe walls and hence was actually utilized in warming the house. As shown in Fig. 28, about 16½ heat units out of the 30 in the flue gases at the smoke collar during the on-periods were utilized, and 13½ wasted. Of the so-called off-periods flue losses and radiation losses from boiler and piping, about 14½ were utilized, and 10½ wasted. About 16 useful units of heat were supplied by sources other than the heating system. Thus, out of every 100 units supplied to the boiler, the following were utilized to warm the house.

	Units
Radiators	45
Flue gas during on-periods	16½
Radiation losses from boiler and piping	14½
Heat from other sources	16
Total	92

This figure represents an overall efficiency of 92/116, or 79.3%. The overall heating plant efficiency, omitting the "heat from other sources," is 76/100, or 76%. Inasmuch as certain losses are inherent in the operation of a heating system, it appears unlikely that much can be expected in the way of reduced fuel costs resulting from the design of still more efficient heating equipment.

Indirect Heating of Domestic Hot Water

Before the correct size and type of equipment can be installed for the indirect heating of domestic hot water (9, 12), it is necessary to know the output and performance characteristics of the heater and, when used, the performance characteristics of the storage tank and connecting piping. Data of this kind were very meager when tests were first undertaken at the Research Home. The investigation is still in progress, hence, the test data are not complete at this time. The present discussion, therefore, will be confined to tests on an internal heater, located in the house heating boiler and connected to a horizontal, insulated, 30-gal hot water storage tank. These tests had as their principal objectives: (1) the availability of hot water and (2) the cost of heating the hot water.

Fig. 29 shows the layout of the test equipment. The boiler had a net I=B=R rating of 63,000 Btu per hr and a gross I=B=R output of 95,000 Btu per hr, when fired at the rate of

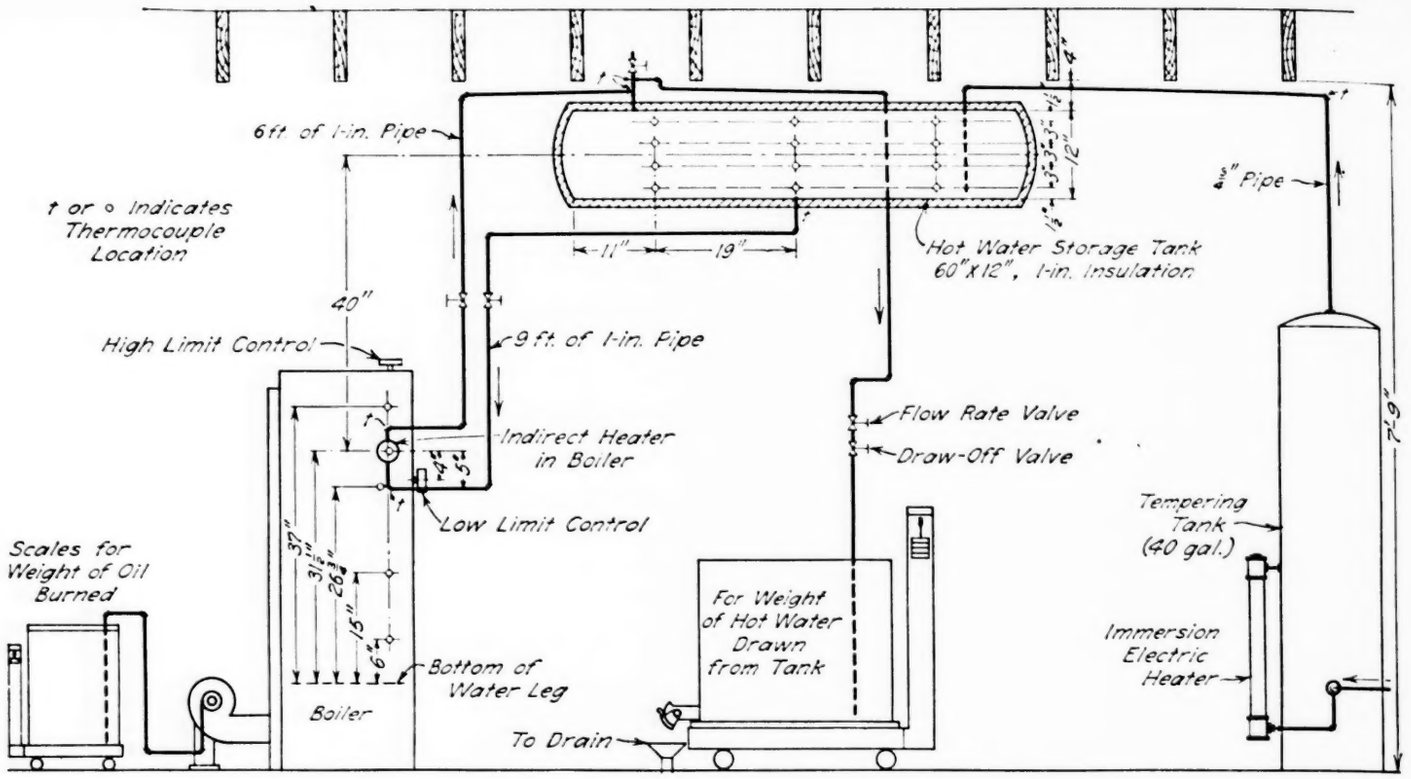

Fig. 29. Diagram of test equipment, domestic water heating.

1.0 gal of oil per hr. The indirect water heater was of the trombone type, consisting of three ¾-inch copper U-tubes that extended into the boiler a distance of 21 inches.

Because space heating is required only during the winter months, whereas domestic hot water is required the year around, it was necessary to run separate sets of tests under winter and summer conditions to obtain complete performance characteristics of the heater.

During the summer months the cost of producing domestic hot water is influenced by the daily fuel consumption chargeable to: (1) heat losses from the boiler; (2) heat losses from the storage tank and piping; and (3) the actual heating of various quantities of hot water. The heat losses from the boiler, storage tank, and piping can be grouped together and termed stand-by losses. To evaluate these stand-by losses two series of tests were run, one with uninsulated 1-inch pipe connecting the water heater and the storage tank, and the other with this pipe insulated. In each of these series, separate tests were conducted to determine the stand-by losses of the boiler, and of the piping and storage tank, at various settings of the low-limit control, ranging from 145F to 210F.

For summer operation, insulating the piping resulted in an approximate saving of 2 lb of oil per 24 hr when no water was drawn off. An additional advantage was realized with the insulating piping, due to the higher average water temperature in the storage tank with a given boiler water temperature. The tests indicated that the boiler water temperature was about 20 degrees higher than the storage tank water temperature when insulated piping was used, and that an increase of 3 to 5 degrees in the boiler water temperature was necessary in order to maintain the same storage tank water temperature when the piping was uninsulated.

It is interesting to note the load imposed on the boiler when a storage

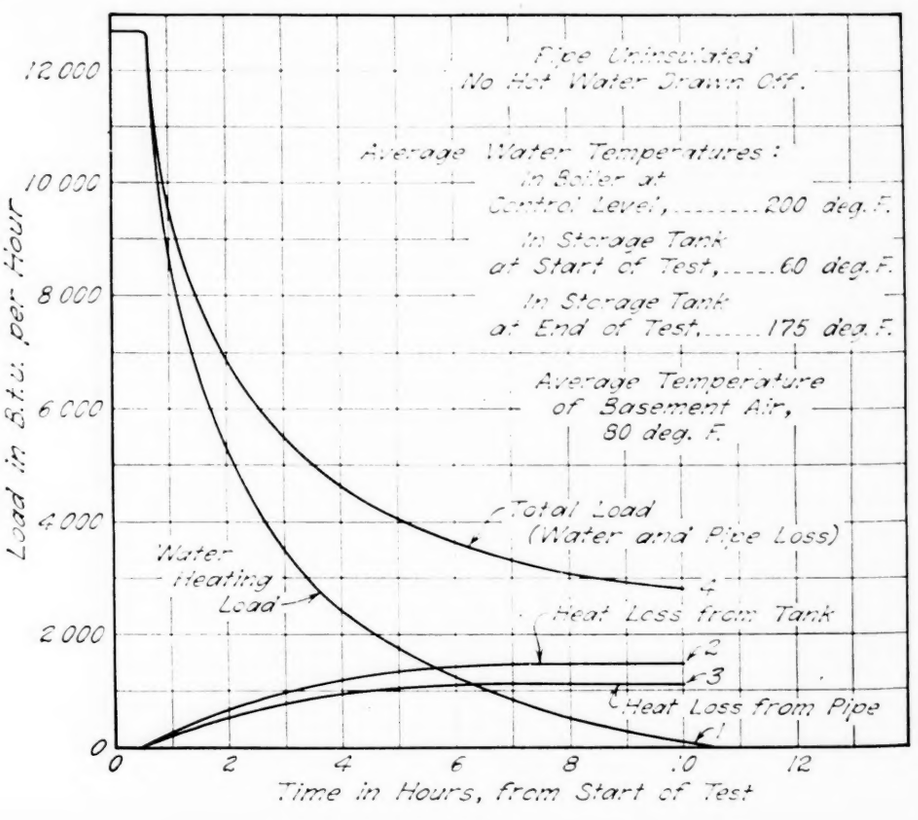

Fig. 30. Water heating load during heating-up period.

tank of cold water was permitted to circulate freely through the water heater immersed in a boiler being maintained at a given temperature.

Fig. 30 illustrates the variation of the load on the boiler with time, for a particular set of initial conditions. It can be seen that the load imposed on the boiler for water heating alone decreases rapidly as the temperature of the water in the storage tank increases. The heat loss from the tank and piping increased as the temperature of the water in the storage tank increased. Tests for other initial conditions indicated a similar relation between the boiler load and time.

Availability

When the water in the storage tank had been heated to a given temperature, there was a given amount of water that could be drawn from the tank before the water was cooled below a usable temperature. Fig. 31 shows the amount of water available above a given temperature for various initial storage tank water temperatures. If 125F is the required minimum temperature, the dotted curve indicates the percent of the total tank capacity that could be drawn off before this temperature was reached. The rate of draw-off for these tests was approximately 5 gpm. The results have very little relation to the performance of the water heater itself, but are useful in estimating the required tank size when the maximum draw-off is known. This type of test is also useful in comparing the mixing action in storage tanks placed in the horizontal and vertical positions.

In order to obtain data necessary for the prediction of the fuel consumption under service conditions, one series of tests was made in which, at various intervals during the day, quantities of hot water approximating those used in actual service were drawn from the storage tank. The three draw-off schedules used are given in Table 10. It is true that the actual demands encountered in service will deviate from the test schedules used. However, other investigations (17) have shown that, within reasonable limits, the actual schedule of draw-offs has practicaliy no effect on the service efficiency, and hence on the daily fuel consumption, as long as the total quantity of water used during the day remains unchanged.

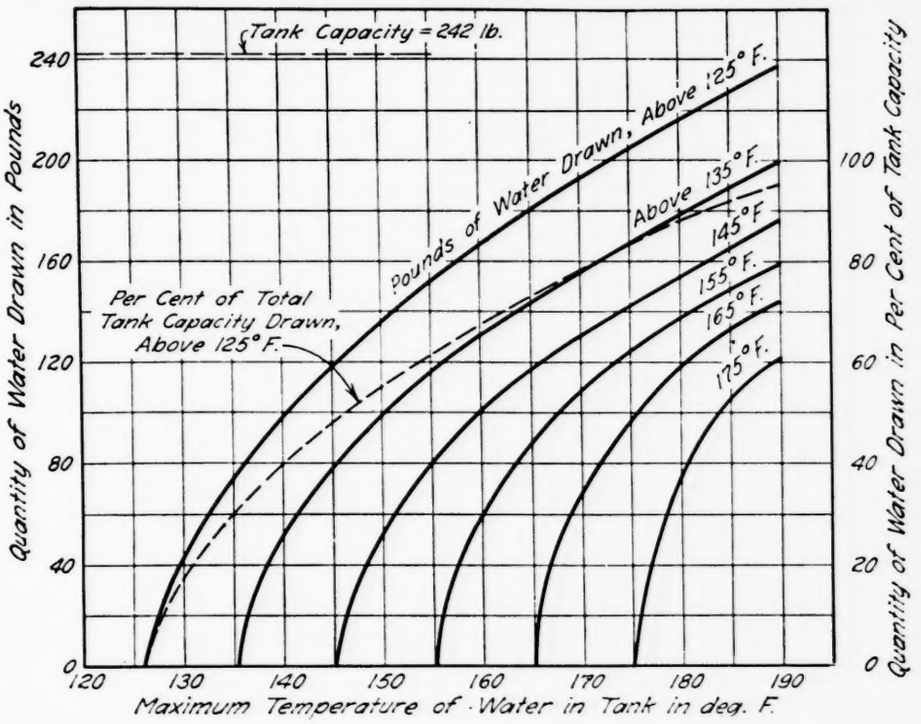

Fig. 31. Hot water available at a single draw-off.

Tests were made at three different boiler water temperatures, using each of the three schedules given in Table 10. Each test was started at the end of a burner on-period, after the system had operated on stand-by sufficiently long to establish steady state conditions. After the last draw-off, as listed on the schedule, the system was allowed to operate under stand-by conditions with no water drawn for the

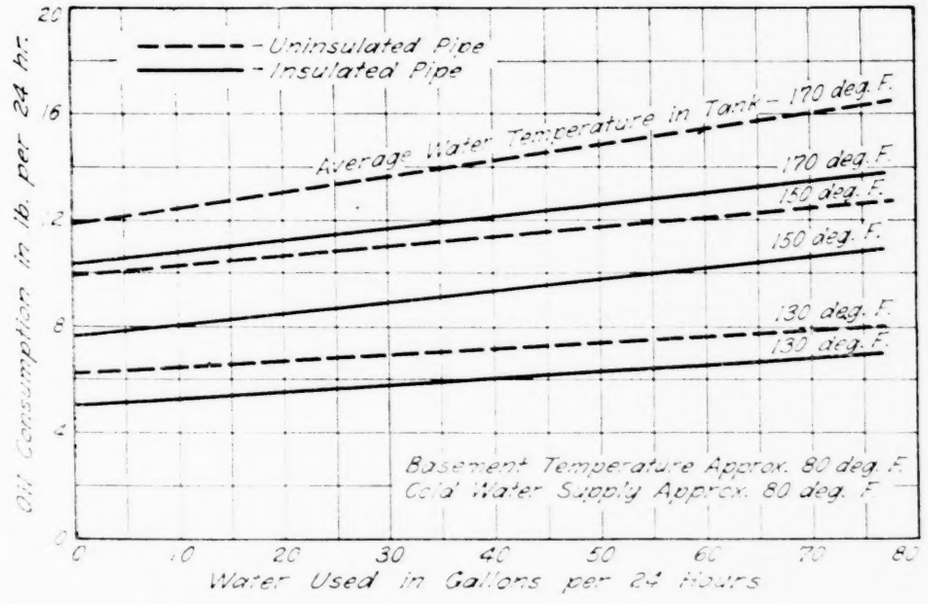

Fig. 32. Daily oil consumption based on quantity of water used—service draw-off tests.

TABLE 10—HOT WATER DRAW-OFF SCHEDULES OF 30, 50, AND 75 GALS PER DAY

Hours from Start of Test	Water Drawn Off, Lb		
	Sched 1	Sched 2	Sched 3
0	17	0	0
1	42	83	83
2	25	58	83
3	8	25	100
4	17	25	42
5	17	25	42
6	17	42	83
7	0	0	0
8	0	0	0
9	0	0	0
10	17	17	25
11	17	25	42
12	25	33	42
13	0	0	0
14	50	83	83
Total lb	252	416	625
Total gal	30.2	50.0	75.0

Start of draw-off test—at end of last burner on-period.

End of draw-off test—at end of last burner on-period, approximately 24 hr after start of test.

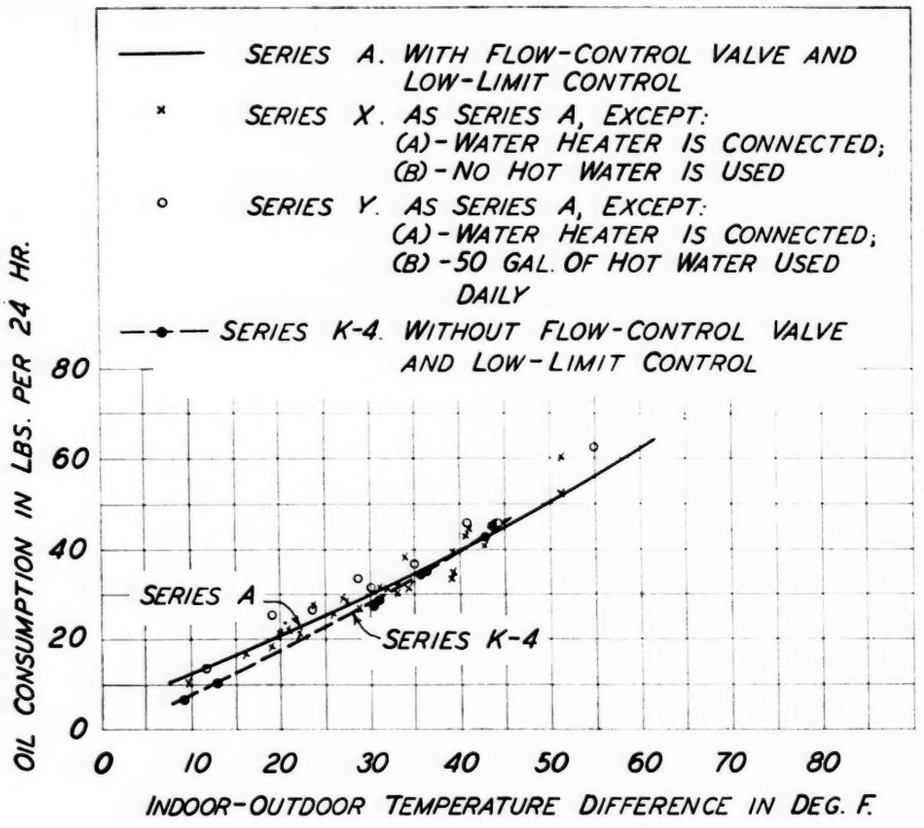

Fig. 33. Daily oil consumption for both heat and hot water.

remainder of the 24-hour period, at which time the test was concluded at the end of a burner on-period.

Comparison between the fuel consumption required to supply a given quantity of hot water when uninsulated piping is used between the water heater and tank, and that required with insulated piping, can be made only when the water in the storage tank is maintained at the same temperature in both cases. Since the operation of the burner was regulated by a temperature control located in the boiler, the daily fuel consumption for each test was plotted against the average temperature of the water in the storage tank, and these curves were then used in connection with the three draw-off schedules to determine the daily fuel consumption that would have been obtained for any given tank temperature. The results, showing the daily fuel consumption at three different water temperatures and for all daily draw-off rates from 0 to 75 gal per day, when operating both with and without insulation on the piping connecting the water to the storage tank, are presented in Fig. 32.

From the curves of Fig. 32 it may be observed that the daily oil consumption chargeable to heating water was reduced by amounts ranging from 1 to 2.5 lb per day, by simply insulating the 15 ft of 1-inch pipe connecting the water heater to the storage tank. In other words, under conditions of summer operation, insulating the connecting piping resulted in a 15 to 20% reduction in the fuel consumption required for heating water for domestic uses, and at the same time somewhat reduced the amount of heat escaping into the house from the piping and chimney. In the summer this heat loss from piping and chimney is undesirable in that it raises the temperature in the living quarters, but in the winter most of it is regained by being utilized to offset part of the heat loss from the house.

Summer Load

Under summer conditions the water heating load is the only load on the boiler, and the only heat that can be regarded as useful is that portion in the fuel burned which is utilized in raising the temperature of the hot water consumed. During the winter the main load on the boiler is the house heating load, and a large portion of the heat that escapes from the surface of the boiler, the chimney, the storage tank, and the piping—which is lost heat as far as summer operation is concerned—is utilized in supplying heat to the house.

In a previous section comparison has already been made of a one-pipe, forced-circulation, hot-water heating system operated: (1) with a control system which would permit the boiler

TABLE 11—DISTRIBUTION OF AVERAGE OUTDOOR TEMPERATURES AND CORRESPONDING FUEL CHARGEABLE TO HEATING WATER

Average Outdoor Temperature, F	Total Number of Days[1]	Average Number of Days per Year	Oil Required to Heat 50 Gal of Water Daily, lb	
			Per Day	Total
1	2	3	4	5
—10 to —5	1	0.2	0	0
—5 to 0	2	0.4	0	0
0 to 5	4	0.8	0	0
5 to 10	11	2.2	0	0
10 to 15	23	4.6	0	0
15 to 20	38	7.6	0	0
20 to 25	68	13.6	0	0
25 to 30	127	25.4	0	0
30 to 35	169	33.8	0	0
35 to 40	150	30.0	1.5	45.0
40 to 45	117	23.4	2.0	46.8
45 to 50	113	22.6	3.0	67.8
50 to 55	104	20.8	4.2	87.4
55 to 60	99	19.8	5.0	99.0
60 to 65	110	22.0	7.2	158.3
65 to 70	95	19.0	9.0	171.0
70 to 75	68	13.6	9.8	133.3
75 to 80	47	9.4	9.8	92.1
80 to 85	17	3.4	9.8	33.4
85 to 90	2	0.4	9.8	3.9
		273		938.0 lb = 134 gal

[1]Based on records of United States Weather Bureau Station at University of Illinois, Urbana, Illinois. Includes months of January, February, March, April, May, September, October, November, and December, from September, 1936, to May, 1941.

to be used to heat domestic hot water as well as to supply heat to the rooms, and (2) with a control system such that the boiler would supply heat to the rooms only. The internal water heater, however, was not connected to the hot-water storage tank during those tests, hence, the oil consumption did not include any fuel chargeable to the stand-by loss from the storage tank and piping, or to the actual heating of water for domestic uses.

Heat Recovered

Two additional series of tests, series X and series Y, were run, therefore, in which the water heater was in operation, and the control system was essentially the same as that used in the series of tests referred to above in which the control system was such that the boiler could be used to heat domestic hot-water as well as to heat the rooms, series A, Fig. 33. For series X, shown by points marked x in Fig. 33, no domestic hot water was used. For series Y, shown by the points marked o, 50 gal of hot water were drawn each day in accordance with the schedule given in Table 10. It is evident that the curve for series A represents series X and series Y equally well. In other words, practically all of the heat losses from the hot-water storage tank and piping were recovered and utilized in heating the house. Thus, with the boiler used both to supply domestic hot water and to heat the house, the only additional fuel required to supply domestic hot water over that required for house heating alone was that necessary to offset the additional heat loss from the chimney. This additional heat loss from the chimney was brought about by the fact that, in mild weather, in order to heat the domestic hot water it was necessary to maintain the water in the boiler at a higher temperature than that which would have resulted if there had been no flow-control valve and low-limit control. With the latter arrangement, the boiler water would have been permitted to cool during times when there was no heat demand from the room thermostat. This condition is represented by the curve for series K-4.

During the winter months the boiler at the Research Home was operated so that it maintained a temperature of 150F in the storage tank, and on the assumption that a family of four uses 50 gal of hot water per day, the curves of Fig. 33 provide a means for estimating the daily fuel consumption chargeable to heating domestic hot water. If the frequency of occurrence of the different average outdoor temperatures during a heating season are known, the fuel consumption chargeable to heating hot water during the winter months can be determined. Since the curves in Fig. 32 supply the fuel costs chargeable to heating water in the summer months, the yearly costs of heating hot water can be determined.

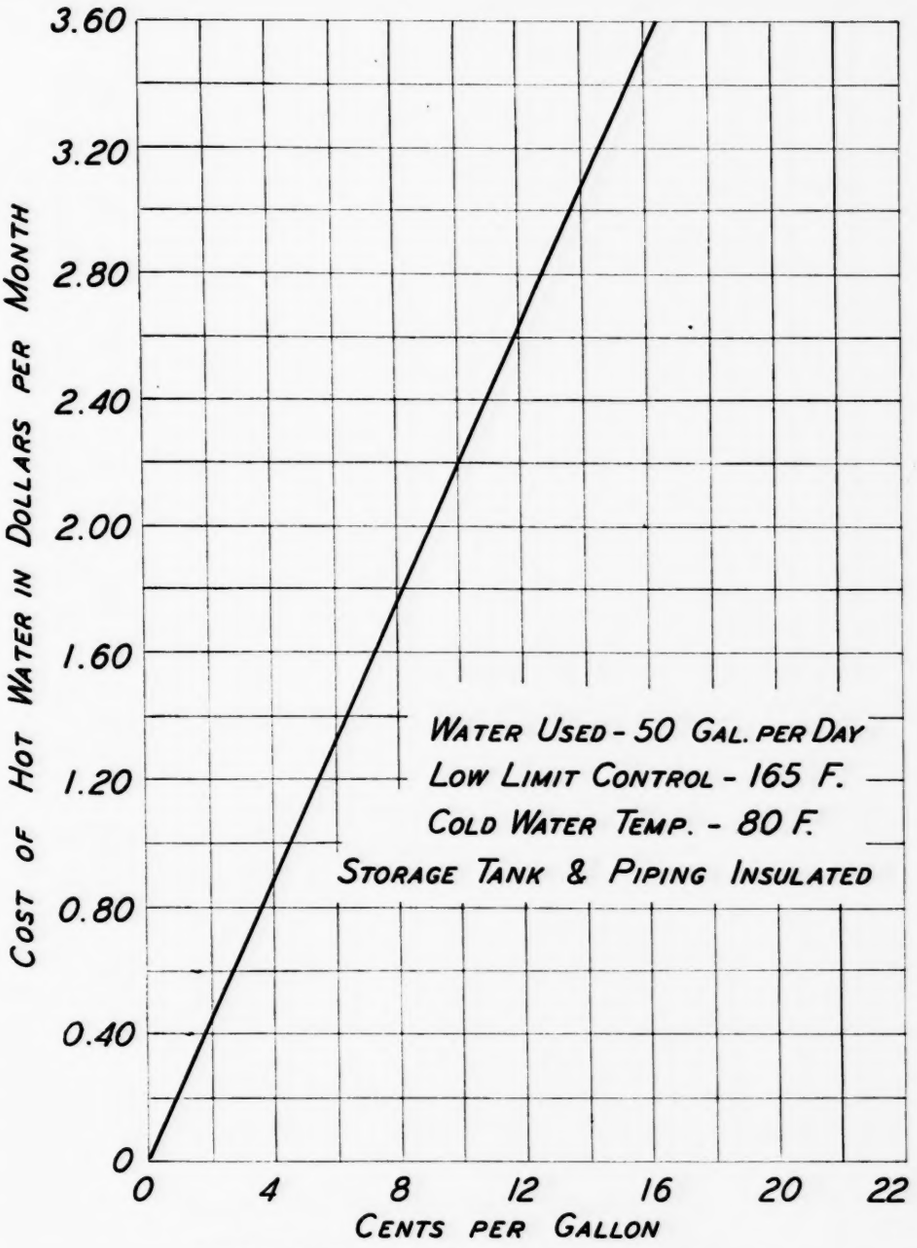

Fig. 34. Average monthly cost of heating water.

The distribution of daily average outdoor temperatures occurring during the nine months of the heating season in Urbana, Illinois, is shown in Table 11. The data for this table were obtained from records of the United States Weather Bureau Station at the University of Illinois, Urbana, Illinois, for the period of September, 1936, to May, 1941, inclusive.

Fuel

Column 4 shows the estimated daily fuel consumption chargeable to heating 50 gal of hot water daily, obtained from the difference between the curves for series K-4 and series Y in Fig. 33, corresponding to indoor-outdoor temperature differences represented by the mean temperature of each bracket shown in Column 1, Table 11. The total fuel consumptions shown in Column 5 were then calculated by multiplying the daily fuel consumptions (Column 4) by the average number of days per year having average temperatures falling within the limits of each bracket (Column 3). The total of Column 5 represents an estimated seasonal oil consumption of 938 lb, or 134 gal for one winter's operation.

Assuming insulated piping, a daily fuel consumption of 9.8 lb of oil per day can be estimated from Fig. 32 for summer operation. The total estimated fuel consumption for the three summer months would therefore be $92 \times 9.8 = 902$ lb, or 129 gal of oil, and the estimated yearly fuel consumption would be $134 + 129 = 263$ gal.

Fig. 34 shows the average monthly cost of fuel for heating 50 gal of hot water per day to 150F with an internal storage type heater connected to a 30-gal storage tank. For the arrangement in the Research Home the cost of power, based on an average cost of 3½ cents per kw-hr for electricity, amounted to approximately 8 cents per month, which should be added to the cost of fuel to obtain total operating cost. For example, if the cost of oil is 13 cents a gallon, the average monthly

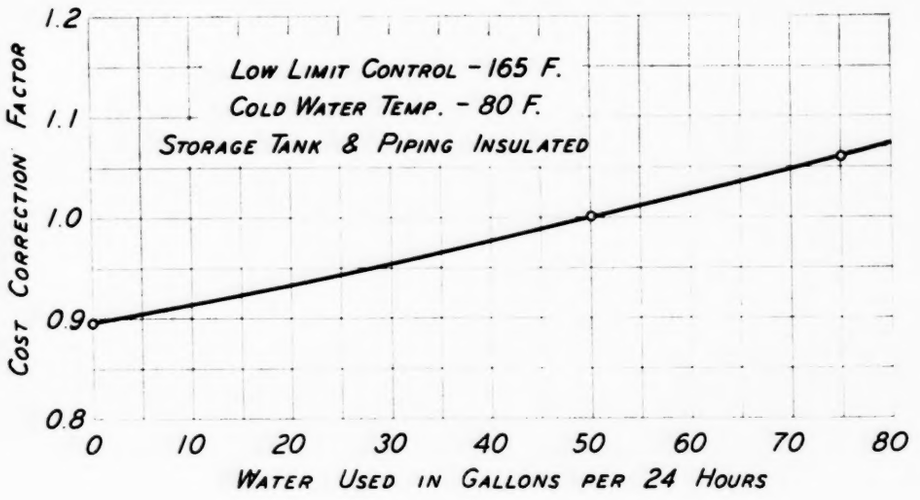

Fig. 35. Fuel cost correction factors for various water usages.

cost of heating water chargeable to fuel consumption would be approximately $2.86. With 8 cents added for the cost of operating the burner, the total monthly cost would amount to $2.94.

Since the fuel consumption curve for series Y (50 gal of hot water drawn per day) was the same as for series X (no hot water drawn), it is quite probable that an increase in the amount of water drawn to 75 gal would still result in the same winter fuel consumption as when no hot water was used. On this assumption the only difference in fuel consumption chargeable to heating hot water would occur in the three summer months. The fuel cost correction factors presented in Fig. 35 were prepared on this basis. For the water usage of 75 gal per day, the curve in Fig. 35 would yield a correction factor 1.06. Thus the average monthly fuel cost chargeable to supplying 75 gal of 150F water per day would be $2.86 × 1.06 = $3.03, a 6% increase.

It is necessary to analyze the data taken in the Research Home on the basis of water in the storage tank at 150F. However, by some agencies it is considered that a temperature of 140F is sufficiently high for all practical purposes. In the latter case, most of the reduction in the cost of heating water would be brought about by conditions prevailing during the summer months.

Data on winter operation were not sufficient to make a complete analysis of the costs with storage tank temperatures of 140F. However, it can be shown from the curves in Fig. 32 that a saving of approximately 18% can be effected in the summer by using water at a temperature of 140F instead of 150F. Since the winter savings would be materially less, an estimated yearly savings of approximately 10% seems reasonable. On this basis an average monthly cost of approximately $2.65 might be expected if water at a temperature of 140F were used and 50 gal drawn off daily.

Economical operation of an indirect water heater is dependent on the use of the proper control system. A very common control hook-up used on hot water heating systems is to have the thermostat control the operation of the circulator, and the temperature control in the boiler control the operation of the burner. This arrangement is simple but it has the disadvantage in resulting in high operating costs unless the operator is careful to keep the limit control setting adjusted in accordance with outdoor temperatures. A control system much preferred to the one just described is one in which the thermostat controls the operation of both the burner and the circulator. This control system is the one used in the tests previously discussed and is described in the section entitled "Control System for One-Pipe, Forced-Circulation Hot-Water Systems." The low-limit control can start the burner but not the circulator whenever the temperature of the water in the boiler becomes too low to be effective in heating domestic hot water, and the high-limit control acts purely as a safety to stop the burner but not the circulator whenever the temperature of the water in the boiler gets too high. This latter method of control may sound much more complicated than the former method but actually it is just as simple to install.

If this system of control, in which the thermostat controls the operation of both the burner and the circulator, is used, the boiler water temperatures will automatically be modulated from a low in summer and mild winter weather, corresponding to the setting of the low-limit control, to a high in very cold weather corresponding to the setting of the high-limit control. Between these limits the water temperature will be just right to properly heat the house with maximum efficiency. With this system of control there will be no need of the operator changing the control settings with change in seasons. Tests indicate that the fuel consumption, when using this control system, will be as much as 20% less than the fuel consumption required with the control system in which the thermostat controls the operation of the circulator only.

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Sivyer Steel Castings Co., Milwaukee, Wis.

Fig. 1. Booth type hood construction for swing frame grinders has proven the most effective method for dust control.

Scrapbook of Exhaust Hood Design

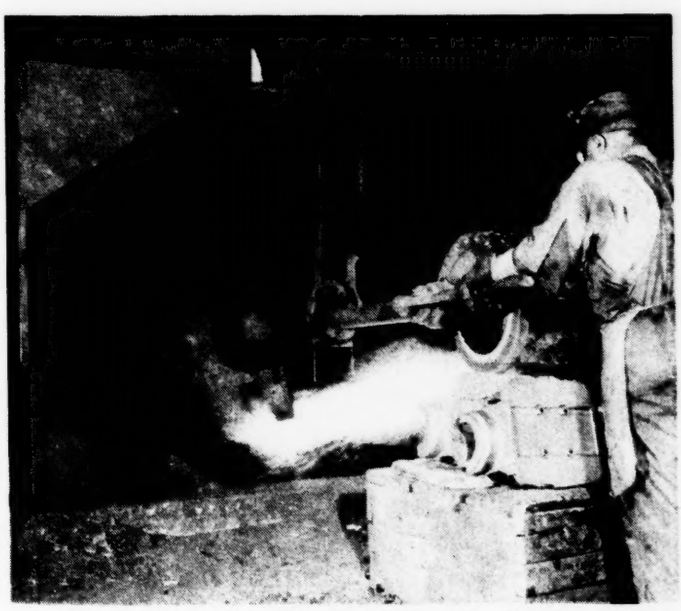
Part 4—Dust Control from Swing Frame Grinders

JOHN M. KANE

Chief Engineer, Dust Control Division, American Air Filter Co., Louisville, Ky.

SWING frame grinding releases heavier loadings of both fine metal dust and coarse molten particles than any other type of metallic grinding. Conventional exhaust hooding at the wheel is impossible due to the movement of the grinding wheel, fusing effect of the heavy stream of molten particles, and the interference of flexible ducts of needed size.

The booth type enclosure, Fig. 1, has proved very

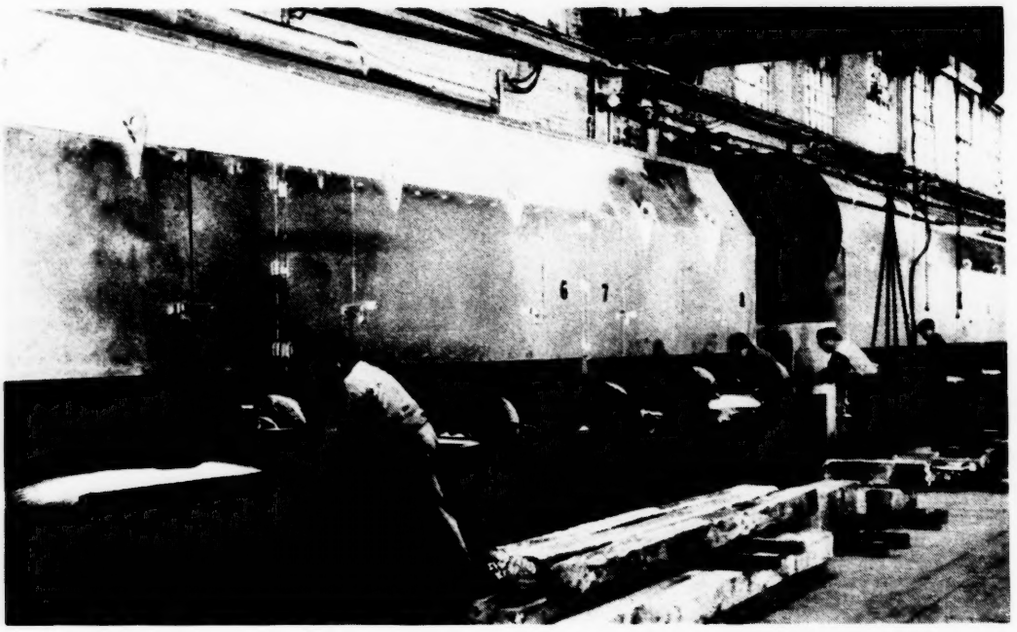


Ross-Meehan Foundries, Chattanooga, Tenn.

Fig. 2. Even where swing frame grinder is several feet in front of booth opening, good control can be obtained at reasonable exhaust volume.

Taylor and Penn, Hartford, Conn.

Fig. 3. Where extreme movement of the grinder is required, location of grinders inside the booth type of enclosure should be given consideration.



Bethlehem Steel Corp., Bethlehem, Pa.

Fig. 4. For billet grinding, hood opening can be modified due to the uniformity of parts ground and ability to keep billets close to booth opening.

effective for a large percentage of swing frame operations. Control is excellent with indraft velocities of 100 to 150 fpm where hood opening is in the order of 25 sq ft or more. A heavy stream of coarse particles is directed into the booth opening and the general air flow toward the opening is ample to capture the fine floating particles even when the point of grinding is 3 to 4 ft in front of the hood opening.

By increasing the face velocity to 200 fpm, good control can be obtained, as indicated in Fig. 2, where the working zone is 6 to 8 ft in front of hood openings. Booth will normally house jib crane or monorail support for the swing frame grinder, although the roof can be slotted for chain fall where monorail is at a considerable height.

Where cast iron parts are finished, cutting speed is so rapid that 360 deg movement of the grinder may be required. In such cases, the enclosure of the operation in a booth, Fig. 3, is necessary for good control with velocities of 100 to 125 fpm over the booth cross-section.

Modification of the booth type enclosure gives excellent results in billet grinding, Fig. 4, where a continuous horizontal opening 24 to 30 inches high is provided and where an indraft of approximately 200 fpm is maintained.

Development Type Homes are Air Conditioned

One of the first medium-priced housing developments in a northern area of the country to offer complete year-round air conditioning is now nearing completion in the Mount Airy section of Philadelphia, Pa.

Built by Max Chipin & Sons, each of the 54 homes in the development is equipped with a G-E packaged air conditioner and a G-E warm-air furnace. While these installations employ gas-fired furnaces, oil-fired furnaces are equally adaptable.

The system was designed and installed by the George C. Lewis Co., Philadelphia, Pa.

The self-contained air conditioner and the furnace are located side by side in the basement utility room, and cool or warm air is carried to the rooms by a common ductwork system. A single high sidewall supply serves each room, with the return located low on the same wall. The change from heating to cooling and vice versa is accomplished simply by throwing a switch. A simple damper arrangement permits diversion of more cool air, if desired on extremely hot days, to the rooms actually in use.

Units installed in basement of house.→

PRODUCT APPLICATIONS

New, Unusual, or Ingenious Uses of Equipment and Materials

Packaged Units Handle Varying Load

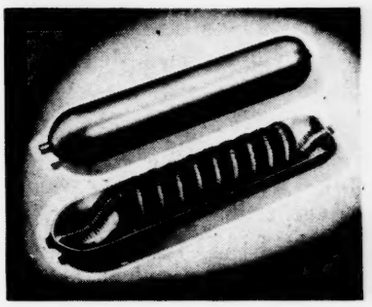
PROBLEM: How to provide sufficient air conditioning for two separate floors with highly variable heat loads was problem confronting the French American Club in Lawrence, Mass. An upstairs dance floor with a capacity of over 400 was often partly or wholly empty, while the ground floor with its bar, pool tables, and card tables could be expected to hold anywhere from 25 to 300 people, depending on the occasion.

SOLUTION: Packaged Typhoon air conditioning units were so arranged that any variation of heat load could be accommodated. On the ground floor, two 10-ton units were installed, and one such unit was located on the second floor. A special plenum extension was built on top of the two downstairs units leading up to the second floor. A damper above the downstairs plenum prevents cool air from being supplied to the floor above except when needed. When the upstairs dance hall requires cooling, the damper is turned by a lever. When the dance floor is crowded and requires more than 20 tons of cooling, the upstairs unit is placed in service. When cooling is diverted to the second floor to accommodate a large crowd, the ground floor is cooled by return air from the upper floors.

RESULT: Bob Lebowitz of Lebowitz Brothers, Typhoon distributors of Lawrence, has followed the occupancy pattern of the club since installing the units and reports that the system works very well. On an occasion when about 400 wedding guests were using the upstairs hall and 75 people were using the downstairs bar, no one complained of discomfort and temperatures were even throughout the club in spite of the fact that all cooling was directed initially toward the upper floor. Another time when there was a small party upstairs and the lower floor was crowded, only the one 10-ton unit was used upstairs and the other two cooled the lower level.

Heat Exchanger Redesigned with Spun Tube

PROBLEM: To redesign a heat exchanger to effect greater compactness without sacrificing efficiency, finned tube was required to be coiled on extremely tight radii and enclosed within a smooth surfaced shell. Previous fabrication methods required four different units: a steel tube shell, two end caps, and a finned tube. There were seven operations involved in the old way: (1) holes punched in shell, (2) shell punched in two caps, (3) finned tube coiled on 4½ inch centers, (4) coil centered, cut, and tails stripped, (5) coil assembled into shell, (6) two end caps assembled, (7) caps welded in position.



SOLUTION: A predetermined length of Wolverine Trufin was coiled on two-inch centers (a bending radius made feasible by the fact that fins are integral with the tube) and inserted in a copper shell which then had both ends spun to the required formed shapes by the spun end process developed by Wolverine Tube Division.

RESULT: Only two units were needed for the new method: seamless copper tube and finned tube. In the new way, operations were reduced to five: (1) holes were punched in shell, (2) finned tube coiled on 2-inch centers without supporting tools, (3) coil was centered, cut, and tails stripped, (4) coil assembled into shell, (5) shell closed at both ends by spun end process. Greater compactness was achieved. Substantial reductions were attained in material and fabrication costs. Marked increase in operating efficiency of the part was realized. These gains were directly attributable to: fewer parts, fewer operations, no shell joints (reducing possibility of leakage), better gas distribution, more efficient use of available condensing surface, and the advantage of heat transfer and durability of copper.

Ceiling Recessed Near Wall to Induce Circulation

PROBLEM: In the new Toledo Union Station, greater air circulation and more daylight than would be expected at the specified ceiling height are required.

SOLUTION: The Fiberglas acoustical tile ceiling is recessed near the glass block wall to induce greater air circulation and afford more daylight. Vents near ceiling level aid in circulation of air which enters through lower windows.

RESULT: Added area of glass block increases daylight, while vents and increased ceiling height promote air circulation.

Traps Hold Seal in Refrigerant Receiver

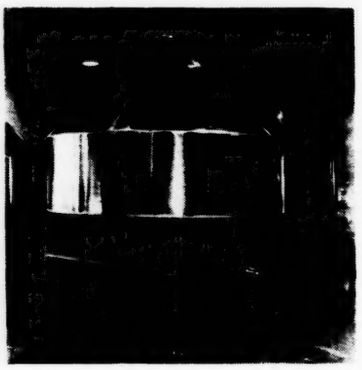
PROBLEM: A problem confronting many refrigeration plants is that of maintaining an adequate liquid seal in refrigerant receivers. If the seal drops below the maximum 6-inch level, high pressure gas blows through the liquid line to the evaporator. Power used to compress this gas is thus wasted. In addition, this high pressure gas adversely affects evaporator operation.

SOLUTION: Eight Armstrong inverted bucket steam traps were installed between the receiver and expansion valve at the Nueva Fabrico de Hielo in Havana, Cuba. During normal operation, with adequate liquid in the receiver, traps are completely filled with liquid. The bucket in each trap is down, holding the discharge valve wide open. When the liquid level in the receiver drops too low, gas enters the liquid line and displaces liquid from the bucket, causing it to float and close the valve. Gas is stopped at this point until the proper liquid level is restored.

RESULT: The eight No. 216 Armstrong traps in this installation pass 500 lb of liquid ammonia per minute. The seals have sufficient power to open at the full differential pressure between the high side and low side of the system. The seals are wide open 99% of the time and do nothing but stand by. The other 1%, according to Nueva Fabrico de Hielo, the traps head off trouble and are an excellent means of cutting needless cost.

Inert Arc Welds Aluminum Without Flux

PROBLEM: The Day Company of Minneapolis, fabricators of steel and alloy products for food and chemical industries, were required to fabricate aluminum salt bins in an over-all height of 10 ft 3 inches and diameters ranging from 6 to 8 feet. Removal of flux and corrosive action on aluminum to entrapped flux was a considerable item in the expense of production.



SOLUTION: The General Electric inert arc welding process was used without the use of flux or special cleaning.

RESULT: According to Verner Holstrom, superintendent of Day Company, the inert arc welder eliminated the expensive flux removal problem and nullified corrosive action due to entrapped flux.

Variable Pitch Sheaves Modulate Heater Output

PROBLEM: For year-round operation of combination direct-fired warm air heating and air conditioning or ventilating systems, variable fan speeds are required.

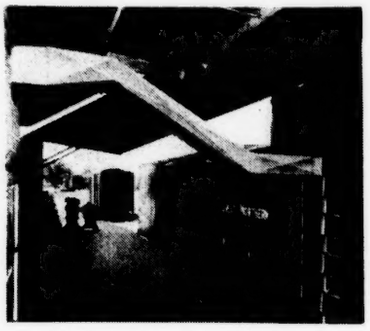
SOLUTION: In one installation for a retail store, an oil-fired Dravo heater of 1,500,000 Btu per hr capacity was used. The 20 hp motor and blower were equipped with variable pitch sheaves, each with eight grooves for belts. During the winter, the blower supplies 17,000 cfm to the heater. The air is passed over the heater's combustion chamber and discharged into the main warm air supply duct. For summer operation, in conjunction with air conditioning, the pitch of the sheaves is adjusted by a lever to increase the blower capacity to 22,000 cfm of cooled air. Variable pitch sheaves can be installed for automatic as well as manual control.

RESULT: Dravo engineers report that installations of this type have resulted in considerable savings due to low initial cost of equipment and the double duty it performs. Wide range of blower speeds also helps balance static pressures and velocities in duct work systems. Maintenance is said to be simplified.

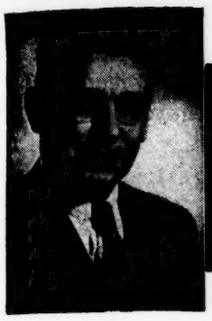
Unit Heaters Heat Box Cars While Loading

PROBLEM: The shipping department of a manufacturing plant required heat in the interior of box cars during the winter loading operations. Cars were loaded from an outdoor platform alongside a heated plant.

SOLUTION: A Modine horizontal delivery unit heater was installed in the wall of the shipping department with a metal duct connected to the face of the unit heater and projected through the outside wall across the loading platform to a point where cars are spotted for loading. During the winter a removable vestibule extends from the building outward to about one foot from the edge of the loading platform. Open spaces between the vestibule and the box car are covered with canvas. In very severe climates it may be necessary to insulate the duct.



RESULT: Heat from the shipping department is usually adequate to warm the passageway enclosed by the vestibule, while heat from the unit heater is delivered directly into the box car, providing comfortable working conditions even in the coldest weather.



Experience is my Classroom

By T. W. REYNOLDS

• DESIGNING FOR THE FUTURE

One of the questions frequently asked by younger men is just what determined this or that design. They visualize a Utopian condition where one sits down and weighs each item of design, discarding this and selecting that, until finally there emerges a flawless concept. But no, that rosy dream vanishes under practical considerations. For example, a house may be so low in cost that only a pipeless furnace is consistent; or in another case the lot may be so oddly shaped as to produce a horizontal arrangement of pipes which is far from desirable.

There comes to mind a pipe tunnel running the length of the basement in a major structure undergoing extensive alterations. Why was the tunnel made as wide as it was? To answer this, one must say that the plans showed it to be much narrower, but being a trench most economically dug by a steam shovel (as later determined by the contractor), it had to be wide enough to suit the workings of his particular shovel.

Still another condition arose on account of the basement having been backfilled years ago with large boulders. Excavations disturbed the boulders and banks caved in, while filling of voids required backfill to be laboriously brought into the existing building. At this point, however, it was found more desirable to widen the tunnel to suit, thus in a sense using concrete as backfill and producing a more commodious tunnel, which, in the end, like all such things, proved none too big for requirements.

In an existing large building with many diverse utilities, the computing of concurrent boiler loads may be in the main a guess, but why worry over it, for conditions are probably already predestined. There is just so much space left which, in all good judgment, one finds more than sufficient, but not sufficient ever to add another boiler unit in the future. So further refinement of the load is foolish and one just puts into the remaining space all of the boiler he can get in, knowing that perhaps he has a boiler slightly larger than he now needs, but nevertheless one that will be most economical in operation and allow for future contingencies now unforeseen. The additional capacity in such a boiler will be most cheaply bought, since the larger the unit the cheaper the cost per horsepower of boiler.

With this boiler, one reflects on what size of equipment he will use for auxiliaries, such as feed-water heater and pumps, though soon deciding that not much figuring is required, since the auxiliaries should be equal to boiler output, even though the output may be in slight excess. No sense in having auxiliaries which cannot keep up with the main item, so that is that.

Perhaps there will be little hesitation on what to do when one comes to the only space inside the building where fuel oil tanks can go. The space, let us say, cannot be extended because a buttress is found supporting the side of a neighboring building, so you take what space you have and decide to receive oil deliveries more often.

All through design one is usually governed by at least two things, two factors of safety which make refinements of calculations unnecessary. One is the provision for emergencies in the way of spares, the other is provision for the future. If there is a spare, one can always operate with this during peaks where the load is underestimated. As for the future, consider a well-water line for air conditioning on whose size you are hesitating. The load is a trifle uncertain, shall the line be 8-inch or 10-inch size, and you finally make it neither, but rather 12 inches for the future. Labor being the chief item, the increased size does not affect the cost too greatly and until the future arrives, the pumping through it will be more economical. If we have underestimated the future, we will have, in this case at least, an ample size for the present.

• ADJUSTMENT OFTEN THE ANSWER

Whenever there are difficulties on an air conditioning job, there is a natural tendency to attribute this to failure of design or equipment. In this way, one seemingly and quickly disposes of the problem with its annoying thinking and labor, but we may often get somewhere faster if the thinking is along the lines of operation, adjustment (tedious though this may be), and a study and analysis of the system. Those who operate the system can best get the feel of it.

I have two recent cases in mind where in each the difficulty was not due to lack of design or adequate equipment, but rather to supply air being at too high a temperature. In one case, heat gain to the ducts was abnormal due to failure to adjust dampers to the point where the area would receive an adequate supply of air. More air up to the designed quantity and there would have been less temperature rise of air within the ducts as well as greater cooling.

In the other case, the high temperature was due to too much air. Less air and there would have been a lower temperature of supply air. Although the greater air movement in the conditioned area (in this case caused by excess fan capacity) was beneficial, it was not as desirable as would have been a lower temperature with the designed air quantity.

It is always best to have colder air, since such air

will drop more readily and have a lower effective temperature around the occupants. Where the excess fan capacity can be eliminated, the air will not only be colder, but also more dehumidified, and here again we will have a lower effective temperature. The efficiency of the coil will only slightly decrease when the air passing over it is less in quantity or velocity.

• FEW OR MANY OUTLETS

In general, distribution of conditioned air is best effected by using more outlets of smaller size rather than few of large size. Cost for outlets and branch ducts is greater, but why jeopardize the satisfactory working of the rest of the system at the delivery end. Then, too, the more outlets there are, the easier it is to correct difficulties, and the less the time taken for total adjustments. Even though there are more outlets there will be fewer of them requiring adjustment.

As opposed to this, there seems to be times in certain small areas when two ceiling outlets might be better than twice that number. It seems that whenever the air to a certain area is low in quantity, it happens to be in a small area with small multiple outlets, and the air just doesn't come down to the occupied space as it does with a large outlet. The delivery from a large outlet seems to have more mass to bring the air down. Possibly this is imagination, although I do know of a couple of installations wherein conditions were corrected by substituting a couple of large outlets for four small outlets.

• NOISY OIL BURNERS

A domestic oil burner installation may operate in a quiet manner, then again it may not. Locating and correcting noise is often tricky, and sometimes the noise disappears as mysteriously as it came. An installation with a single oil tank may run for years without noise, then when a second tank is added there may be a so-called tank hum in the second tank.

In a recent case of this kind, no hum was ever heard in the first tank, because it was laid horizontal and also rested directly on the concrete floor. With the addition of the second tank a hum was produced in the second tank since, according to modern and proper installation practice, this tank was set on short uprights off the floor and was not laid flat. Slight throttling of the return line valve at this tank minimized the noise, but only when the tank was nearly full.

A single cure may eliminate a tank hum in some cases but not in others; it depends on the particular installation. A simple method, and one that has always corrected the difficulty in the writer's experience (although some say it will not always do the trick), has been the forming of four small loops in the return pipe at its connection to the tank. These loops then take up the shock or vibration within the line. The line itself should, of course, be kept from contacting the tank except at its connection to the tank.

Some burners are furnished with an anti-hum valve to be installed only if required. This may be a ball-type valve having the ball held on its seat by a coil spring, the tension of which may be adjusted by hand wheel so that the pump will maintain a vacuum of from 2 to 10 in. Such a setting must be made with a vacuum gage, otherwise the pump may fail to pick up oil quickly after a long idle period.

Other burners may have a small restricting screw or bushing screwed into the top of the strainer assembly. In this bushing is a hole sized to restrict the oil flow to pump suction, so that a fairly high vacuum will be maintained. If the flow is restricted too much, the oil flame may become erratic, both in starting and while running.

Still another anti-hum valve consists of a flexible diaphragm under the strainer cap to absorb the vibration in the column of oil, yet without interference with the flow of oil. Some service men say that an anti-hum valve is not always the cure. The experts vary in opinions according to their experience.

A boiler or furnace with a high draft loss or chimney of inadequate size will restrict the gas flow at high firing rates. The gases then have difficulty in getting out of the combustion chamber fast enough. The result is a rumbling or pulsating sound which can be eliminated by reducing the size of the fire. Since this is a combustion noise, the next smaller size of burner nozzle reduces the amount of combustion and is therefore often the answer. The burner will then run slightly longer to handle peak loads, but economy will result due to lower stack temperature.

Another remedy is to crack the fire door or otherwise provide a small opening of about one inch in diameter (too large an opening will lower the CO_2). Increase in height of chimney, an induced draft blower, or mere cleaning of restricted flues may also be the answer to combustion noise.

• WHEN NOT TO USE AN OUTLET

Far be it from me to disparage in the slightest the desirability of grilles or other form of supply air outlets, so it is with some hesitation that I say I have known of a few rare cases where removal of the distributing devices has actually improved defective cooling conditions. One should not design that way, but can sometimes correct a condition in that way where coil or refrigeration is under capacity. For example, removal of the outlet removes quite a bit of resistance, particularly where the total supply air quantity has to be stepped up. Removal of the resistance, of course, increases the air velocity over the coil, hence the amount of refrigeration. Less outdoor air will also be taken in through the outdoor air damper, thus lightening the refrigeration load.

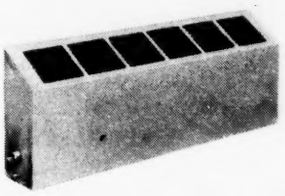
A recent installation, but with reversed situation, brings the subject to mind. This air conditioning installation is in a restaurant where the space was very cool for a short time and during that period quite a relief to the short-term occupant. Pending receipt of the various grilles and outlets, the new system was run and the air dumped out on the occupants without any form of grille or outlet. Once these were installed, the cooling effect became noticeably reduced, although still satisfactory.

NEWS OF EQUIPMENT AND MATERIALS

Use the prepaid postcard opposite Page 118 for securing additional information on items described in this department.

Modine Adds Two Convectors

Two convertors have been added to the Modine line by the Modine Mfg. Co., Racine, Wis. The Type F-12 has an over-all height of only 12 inches and is furnished for free-standing or recessed installation under picture windows. It is available in 6- and 8-inch depths, and in 8 lengths ranging from 40 to 96 inches.



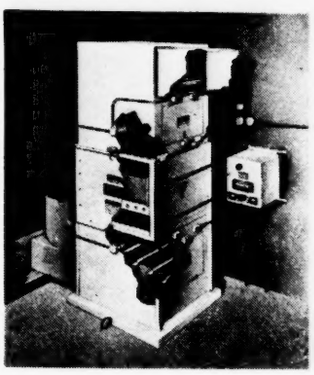
Enclosures longer than 72 inches are designed with removable center supporting legs for greater strength and rigidity. On these lengths, dual dampers and lower grilles (optional) are also provided. Fronts and grilles are removable for full access. Enclosures are proportioned to harmonize with the standard Type F convectors.

The Type S-14 convector (illustrated) is intended for heating bathrooms or other areas where heat loss is small or space is limited. The sloping top wall cabinet is only 11 inches high and 4 inches deep and is available in lengths of 10, 18 and 26 inches with steam ratings of 5, 10, and 15 sq ft, respectively. This convector may be installed as low as 3 inches above the floor. Supply and return connections of the reversible heating unit are at the same end.

For more information, circle on Inquiry Card, **Item 1**

Vertical Air Flow Precipitron

A Precipitron electronic air cleaner, constructed for upward and downward air flow to conserve floor space and simplify maintenance, is now available from Westinghouse Electric Corp., Sturtevant Division, Hyde Park, Mass.



Designed to remove dust and other air-borne solids from normal air, the unit charges the solid particles positively, then passes the air through a set of alternately charged collector plates where the solids are removed from the air by electrostatic attraction.

The Vertical Air Flow Precipitron finds typical use in heating and air conditioning systems, and for oil-mist collection. It is available in ratings of 1,200 cfm and up. It is said to remove 90% of all air-borne particles traveling at 333 fpm, and 85% of those traveling at 400 fpm horizontal air flow. Mounting the unit in a vertical duct results in saving floor space. Unit frame has a projected floor area of 4 sq ft for the smallest unit to 16 sq ft for the largest unit. Height is 34½ inches. Bulletin 93-815 is available.

For more information, circle on Inquiry Card, **Item 2**

Hydraulic Motor Valve

The Sarcostat, an electro-hydraulic motor valve for open-and-shut control, is being produced by Sarco Co., Inc., New York, N. Y. This valve is designed for automatic operation by thermostats or pressurestats or by liquid level or flow controls.

As a shut-off valve in an inaccessible location, it can also be actuated by a push-button. The valve is constructed to handle steam, water, oil, gas or air.

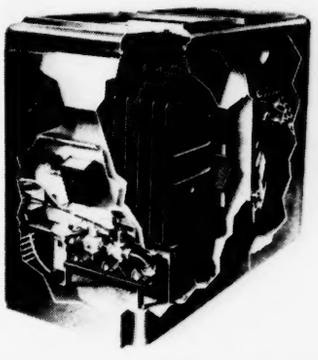
The valve operator, actuated by hydraulic power, operates single seated valves up to 1½-inch or double seated valves up to 4-inch at 125 lb per sq in. It does this by direct thrust, without resort to pilots, gears or levers. When the full stroke is used, standard valves will open in 60 seconds and close in 10 seconds. The unit is also available with an adjustable stop for partial throttling.

Valve bodies are of brass or iron with union or flanged connections. Various types of valve bodies are available: single seated, needle, piston or double seated. Bulletin No. 1080-A is available.

For more information, circle on Inquiry Card, **Item 3**

Compact Gas-Fired Furnaces

The Ingersoll GL series of gas-fired winter air conditioners is announced by Ingersoll Steel Division, Borg-Warner Corp., Chicago, Ill., with input ratings of 78,125, 93,750, 109,375, and 125,000 Btu per hour. All models in the series are compact, due to the inclusion of a wedge-tube heat exchanger.



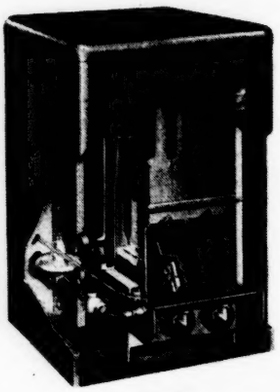
The two smaller low-boy models occupy only 20 x 51 inches of floor space, and the two larger models need only a space of 20 x 54 inches.

These units have AGA approval for use with natural, mixed, manufactured, LP, and LP-air gases. The units are factory assembled and wired. Assembly includes heat exchanger, blower, filters, raised drill port burners, and fully automatic controls.

For more information, circle on Inquiry Card, **Item 4**

Gas-Fired Boilers in 5 Sizes

Expanding its line of residential gas heating equipment to accommodate the heating needs of medium and larger homes, the Timken Silent Automatic Division, Jackson, Mich., announces a new line of gas-fired hot water boilers in 5 different sizes and capacities, from 72,000 to 144,000 Btu per hr at the boiler outlets.

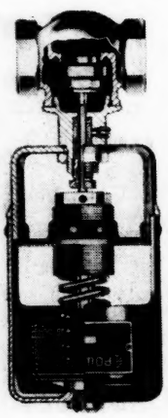


The sectional cast iron boilers are of vertical tubular design and are baffled for maximum heat absorption. They are completely equipped with burners and all necessary controls. The well-proportioned, rectangular cabinet is finished in baked enamel. Unit has a steel base, permitting installation on wood floors, and is thoroughly insulated to minimize heat loss. Because of its compact design, it occupies little floor space.

For more information, circle on Inquiry Card, **Item 5**

Air Operated Control Valve

An air operated control valve for regulating the flow of steam, Freon, water, oil or gases, termed the Meta-flow valve, is being produced by Powers Regulator Co., Chicago, Ill. While the standard model is shown, it is also available with a packless valve body. The diaphragm is operated for pressure differentials below 75 lb per sq in. It is available in sizes from ½ through 2-inch.



High-strength aluminum alloy is used for the housing, and bronze for the valve body. A polished stainless steel stem moves in preformed lubricated metallic packing. The bellows is made of long-life brass. A brass adjusting screw, with steel spring having 15 lb adjustment range, provides proper sequence operation where required.

For more information, circle on Inquiry Card, **Item 6**

Convector With Damper

Certified convectors, in 20- and 24-inch heights, are manufactured by Certified Products, Inc., Newark, N. J., in two models—free-standing, and a model for recessed installations. All units are equipped with pre-installed dampers.

Cabinets are fabricated of heavy gage sheet steel with rounded corners on front and top. Fronts are designed for easy removal and are close-fitted to prevent air leakage.

Heating elements consist of three ⅝-inch OD seamless copper tubes and flat aluminum fins with extruded collars. Tubes are mechanically expanded into the fins. Headers are made with offset tappings to provide ample room for wrenches.

Units are supplied with 2¾-inch tappings for inlet and outlet and 1¼-inch tappings for venting on heating elements used for gravity, forced hot water, and 2-pipe steam systems. For 1-pipe steam installation, tappings are 1¼-inch inlet fittings and 1¼-inch vent fittings. Other tapping arrangements can be obtained.

For more information, circle on Inquiry Card, **Item 7**

Flow Control Has Vent Tube

A Thrush vertical flow control valve with air vent tube, announced by H. A. Thrush & Co., Peru, Ind., is intended to eliminate a pipe nipple and elbow and thus reduce installation cost. It comes in the following sizes: 1, 1¼, 1½ and 2-inch.

The unit is used with automatically fired forced circulating hot water heating systems. The vent tube assures positive air elimination from the top of hot water heating boilers. Air is placed back in the pressure tank where it serves a useful purpose. Valve is also available without the vent tube.

While all four sizes have union outlet connection, the first three sizes can be supplied with the union outlet for either threaded pipe or solder connections.

For more information, circle on Inquiry Card, **Item 8**

Sweat or Solder Check Valve

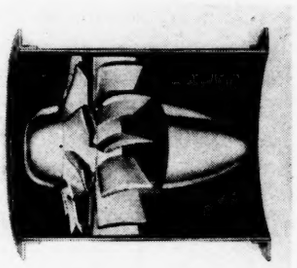
A check or flow valve, available for either sweat or solder type connections and intended to prevent hot boiler water from flowing to a heating system when the circulator is not running so that the boiler can be used for heating domestic hot water, has been announced by Taco Heaters, Inc., Providence 3, R. I.

Valve is of all-bronze construction with accurately machined weight and seat, bronze spring cushion, visible three-position (open, closed and normal) adjustment. The seat is easily cleaned by moving the adjusting lever up and down with the circulator running. The valve is made in one size, 1 inch.

For more information, circle on Inquiry Card, **Item 9**

Vaneaxial Fans — 2 Models

Two new vaneaxial type, medium pressure, propeller fans have been announced by Propellair Division of Robbins & Myers, Inc., Springfield, Ohio. Designed for ductwork applications, the fans permit series or straight-through connections for all air-moving requirements. Guide vanes prevent swirling. A smooth, efficient, axial flow of air is provided. Operating on a low horsepower input, units work quietly over a wide pressure range.



Steeply pitched blades on both the propeller and vane, designed with high lift sections at the root, utilize the latest laminar flow airfoils. Pitch and lift sections are reduced as they approach the tip to provide uniform discharge velocity over the entire blade areas.

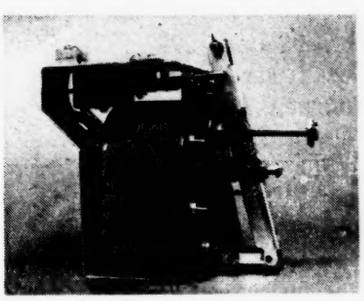
Spun aluminum nose and tail cone are contained within the distance between the two mounting flanges of a 30-inch duct section. Easily installed in standard 20- and 24-inch dia duct systems, units are equipped with open-ventilated, ball bearing, 40C rise, standard NEMA foot-mounted motors.

Type VCDD, direct-driven unit is constructed with the motor inside the airstream, suitable for non-hazardous applications. Type VCS, belt-driven unit is recommended for installations in which corrosive or inflammable fumes or dusts must be considered.

For more information, circle on Inquiry Card, **Item 10**

Control for Trolley Coaches

An automatic temperature controlling system for trolley coaches has been announced by the Minneapolis-Honeywell Regulator Co., Minneapolis, Minn. The units of the system, specially adapted for trolley coaches, are the same as those employed on bus controls, making possible a considerable inventory reduction for transit firms operating both busses and trolley coaches.



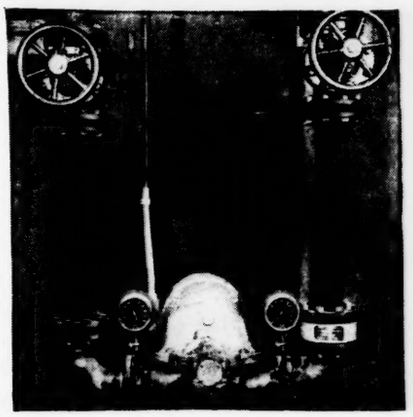
The controls operate on the modulating principle and employ a single pneumatic thermostat operating an air motor (illustrated) connected to the waste heat damper. An end switch is mounted on the motor and turns on auxiliary heat whenever the damper is fully opened.

Constant interior temperatures can be maintained and alternate blasts of heat eliminated through use of the new controls. In operation the thermostat automatically positions the damper to permit a constant flow of heat into the coach at any selected temperature. The heat, created by electrical resistance, which is not needed in the coach is directed to the outside. When all waste heat delivered into the coach is insufficient to maintain selected temperatures, the end switch automatically turns on auxiliary heaters.

For more information, circle on Inquiry Card, **Item 11**

Noise and Vibration Absorber

A new noise and vibration absorber unit, Soundzorber, for installation in pipe lines, is announced by Finn & Co., New York, N. Y.



It is a specially constructed rubber pipe used as a flexible connection between the pump and the pipe line and is said to be ideal for use in water pumps, water systems, boiler systems, and air conditioning systems.

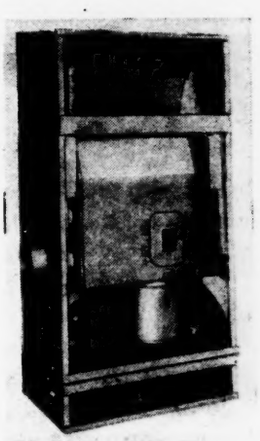
Standard flanges on each end of the pipe are imbedded in the rubber. It is easy to install, just bolt on to flange of the machine. No gaskets are necessary. There is no metal-to-metal contact between the pump and piping system.

Unit is designed for working pressures up to 250 lb per sq in. It is available for pipes from ½ to 14 inches in diameter and will withstand temperatures up to 250F.

For more information, circle on Inquiry Card, **Item 12**

Oil-Fired Wall Furnace

An oil-fired wall furnace with circulating blower fan has been introduced by Delta Heating Corp., Jackson Heights, N. Y., as Model WF-100. Unit includes a front panel and outlet and inlet grille, not shown on accompanying cut-away view.

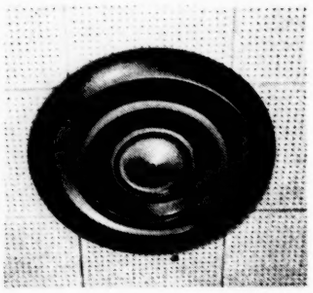


Unit is equipped with a 10-inch Breeze oil burner with a 10-inch pot and has an hourly output of 52,000 Btu. Furnace is equipped with a 4-blade, 10-inch fan and has a casing 60 inches high by 31 inches wide and is adjustable in depth from 18 to 20 inches. The air delivery is 800 cfm. Fan is two-speed. The furnace is equipped with an automatic draft regulator and automatic safety thermal control.

For more information, circle on Inquiry Card, **Item 13**

Two Models of Ceiling Outlets

Designed for installation where the outlet cannot be recessed into the ceiling, the new Barber-Colman Model JS and JSC air distribution outlets are being produced by Barber-Colman Co., Rockford, Ill. The outlets have adjustable air deflection and a high aspiration rate, and extend only a few inches above the ceiling surface.

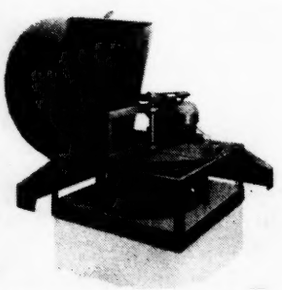


These models are available as supply or combination supply and return outlets. Supply air deflection is adjustable from vertical to horizontal in supply outlets and limited in combination units to prevent short circuiting of supply air to the return opening. Outlets are supplied in a wide range of sizes in a baked metalescent aluminum finish or prime coat for painting.

For more information, circle on Inquiry Card, **Item 14**

Centrifugal Fan Ventilator

Especially suitable for situations where powered duct exhaust ventilators operating at very low noise levels are desirable, the Airlift fan ventilator is announced by The Swartwout Co., Cleveland, Ohio. The power unit features a centrifugal type fan with backwardly-curved blades, mounted within a weatherproof chamber.



Streamlined fan bottom is said to increase efficiency while reducing turbulence to a minimum and permitting slower operating speeds. Welded steel framework provides sturdy support for the unit which fits over curb built to extend duct rising from building roof.

The ventilator is available in 14 sizes, providing nearly 50 capacity variations.

For more information, circle on Inquiry Card, **Item 15**

Device Tests Gas Leaks

The Gauge Rite Air Test has been put on the market by Marbro Mfg. Co., Cleveland, Ohio. It is a small, compact, and lightweight gas line tester for testing gas leaks in ounces.

Use of the device is simple, easy and fast. The pressure is built up by squeezing the small aspirator a few times with the fingers. Because it is compact and lightweight, it can fit into the smallest work areas. It comes complete with a handy metal carrying case. The weight of the unit is 1½ lb, complete with the case. The sensitive gage registers up to 20 ounces, with 10 ounces recommended as the correct test pressure. The valve is spring-loaded, turn, snap-action type. The body has a standard ¾-inch thread. Adaptors can be applied to increase or decrease the pipe size.

For more information, circle on Inquiry Card, **Item 16**

Boiler-Burner for Small Homes

A boiler-burner unit for small homes has been announced by Weil-McLain Co., Michigan City, Ind. Known as the OB-15, it consists of a cast iron boiler designed for oil firing with the Type A Weil-McLain oil burner. It has an I-B-R net connected rating of 235 sq ft of steam and 435 sq ft of water.

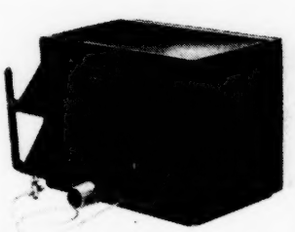
The unit is shipped assembled, and the jacket, insulation and combustion chamber are designed for easy, speedy installation. Boiler is round in design and is available with tankless or tank type indirect water heater, located in the top section for maximum heat transfer. The sturdy steel jacket encloses burner and controls, and is finished in an attractive grey enamel.

Over-all dimensions with jacket are 22½ inches wide, 38 inches long and 41 inches high.

For more information, circle on Inquiry Card, **Item 17**

Large Capacity Gas Burner

For installations requiring a horizontal gas burner, Iron Fireman Mfg. Co., Cleveland, Ohio, announces the GCR series radiant gas burner. There are 18 models in the series.



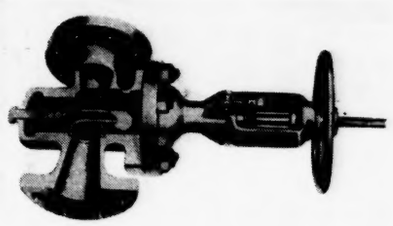
The unit produces a soft flame, shorter than is usual with horizontal burners, for two impinging gas jets are introduced into each of 16 oval tubes in the face of the burner block in such a way that air and gas are rapidly mixed. In addition to primary air carried in through the oval tubes, there are nine secondary air openings in the face of the burner block, one of which may be used for insertion of a pilot light. Illustrated is the rear view of the burner showing the air tubes.

The manifold, placed in line with the refractory or burner block, is so designed that it will not interfere with air entering the oval tubes.

For more information, circle on Inquiry Card, **Item 18**

Ball Valve for Throttling Use

A venturi-ball valve especially designed for throttling applications in high pressure, corrosive or erosive applications, featuring straight line discharge characteristics, has been announced by W-K-M Co., Houston, Tex. In the throttling position, the ball presents a smooth surface, and non-turbulent flow is possible over a wide range without destructive erosion or wire drawing.

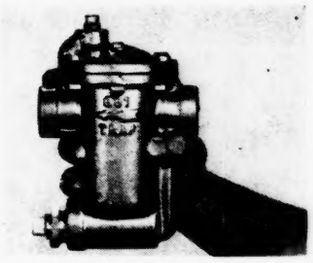


Wide open flow is unobstructed, non-turbulent with negligible pressure drop. When tightly closed, the ball seats tightly regardless of pressure and flow conditions. Line contact between ball and seat assures tight, absolute shut-off. It is self-cleaning, for liquids or gases passing through the venturi passage suck out dirt, scale, sediment. The ball seats in a different position each time. Valve, which has two seats, may be reversed in the line to double its useful life; it is available in different materials. Details are given in Bulletin No. 695.

For more information, circle on Inquiry Card, **Item 19**

Steam Trap Has Strainer

A small inverted bucket steam trap, which has a brass strainer built right into the body, is a product of Armstrong Machine Works, Three Rivers, Mich. Known as No 881, this side-inlet, side-outlet trap is used for draining smaller sized equipment producing relatively small amounts of condensate. The built-in strainer feature eliminates the need for a separate strainer, extra fittings and installation labor.



The trap has ½- or ¾-inch pipe connections; operating pressure 250 lb per sq in.; 769-1,060 lb per hr hot condensate capacity; 7 1/16-inch height; 5-inch diameter, 6 lb weight; cast semi-steel body and cap; 18-8 stainless steel bucket.

For more information, circle on Inquiry Card, **Item 20**

Rubber Pad Shock Kit

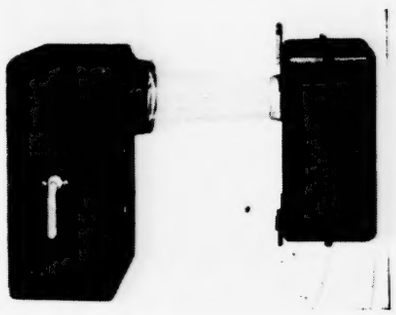
A self-engineered rubber pad kit, that will eliminate up to 75% of the vibration and shock caused by machinery, has been introduced by the B. F. Goodrich Co., Akron, Ohio, as the answer to single problems of industrial vibration too small to justify the use of technical personnel.

Named Vibropad, the kit consists of a 12-inch square rubber pad, rubber tubes, thick washers, and an instruction booklet. It was designed specifically to enable the small plant owner with only two or three machines to install the pads under the legs of the equipment and to save the cost of engineering and installation.

For more information, circle on Inquiry Card, **Item 21**

Smoke Detector for Ducts

A photoelectric smoke detector, Type A28E, for air conditioning duct systems, is announced by Photoswitch Incorporated, Cambridge, Mass. It is recommended for use in stores, theaters, hotels and other locations where the presence of smoke is a hazard to property or a possible cause of panic.



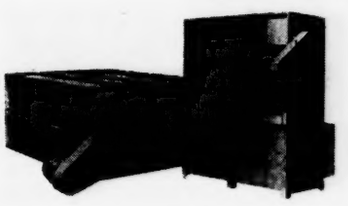
At the first sign of smoke this unit operates, and through auxiliary relays (not included) may automatically turn off blowers, close automatic shutters and signal the maintenance department.

The light source and receiver are designed for mounting on opposite sides of the duct. Operating range is from 5 to 30 ft. Among the features of this system is a sensitive photoelectric circuit with fail-safe circuit design and a tamper-proof sensitivity control to prevent unauthorized adjustments. Bulletin PE 505 is available.

For more information, circle on Inquiry Card, **Item 22**

Sectional Units Cut Costs

The YAC Young Air Conditioning line, produced by Young Radiator Co., Racine, Wis., is claimed to offer greater versatility in application through use of a sectional design. Units are shipped in sections, unless otherwise specified, to provide greater ease in handling, installing and servicing, as well as simplifying specification of this air conditioning line.

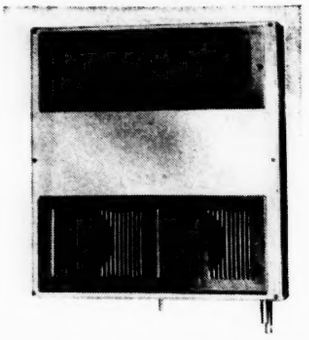


Units, stocked in 8 sizes, horizontal or vertical types, provide any combination of service desired, from simple heating and cooling to complete air conditioning—heating, cooling, humidifying, dehumidifying, and filtering. Capacities range from 400 to 16,625 cfm under standard conditions. Catalog No. 7550 gives dimensional and other data.

For more information, circle on Inquiry Card, **Item 23**

Fan-Convector Heats and Cools

A forced air convector, said to be ideal for many special applications, such as tourist cabins and hard-to-heat rooms, has been introduced by Auburn Burner Co., Auburn, Ind. Each cabinet contains one or two small centrifugal blowers powered by quiet 1/150 hp motors. These force the air over and through the extended surface coils and out the upper grille.



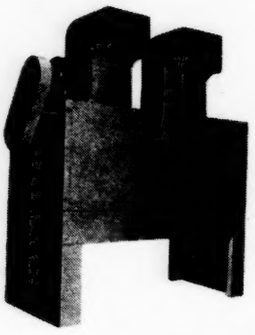
Each convector can be controlled by a separate thermostat if desired. The radiator is equipped with an automatic vent. Cooling can be accomplished in summer by circulating cold water through the coil.

Convector is made in a single blower model with 12,000 Btu per hour capacity, and a dual blower model with a capacity of 20,000 Btu per hour.

For more information, circle on Inquiry Card, **Item 24**

Blower Type Unit Heater

A blower type unit heater for heating large areas is being produced by U. S. Air Conditioning Corp., Minneapolis, Minn. It is manufactured in eight models with capacities from 35,000 to 2,000,000 Btu and 1,200 to 33,000 cfm.



The heater is composed of a basic blower section with the heating coil attached. Blower section can be easily passed through ordinary doors and windows. Outlets can be rotated to face in any direction. The outlet cowls can be attached to the blower outlets in any of four positions—units may be inverted, floor, ceiling or wall mounted.

The basic blower section is constructed of heavy gage steel, with corrosion-resistant finish inside and out. Self-aligning flange type bearings, in an efficient two-bearing design, are mounted outside the housing, away from the heat, allowing easy lubrication. Forward curved blowers make the heater efficient and quiet.

For more information, circle on Inquiry Card, **Item 25**

Wall Circulator Saves Space

A dual vented gas wall circulator, Pion-Aire 50, is announced by Pioneer Furnace Co., Division of Pioneer Water Heater Corp., Los Angeles, Calif., for use in homes, motels and offices. Its capacity is 50,000 Btu per hour.

Unit can be fitted into a standard 2 x 4-inch stud wall with studs spaced 16 inches on center. Installation is accomplished without the need for basement or floor space. In occupied homes, it is not necessary to dismantle and reassemble the entire wall.

Dual heat is provided by a conveniently placed control which permits diverting the warm, circulating air into either adjoining room. A vent carries off products of combustion. Unit operates on natural and manufactured gases.

For more information, circle on Inquiry Card, **Item 26**

Thermostatic Expansion Valve

A non-adjustable thermostatic expansion valve for fractional tonnage refrigeration applications, such as room coolers, is being produced by Automatic Products Co., Milwaukee, Wis., for use with F12, F22, and methyl chloride.

It is possible to clean the Monel strainer without removing outlet or inlet connections. Valve can be finished to meet any superheat setting from 2 to 14 deg. Unit is available with either a short or long thermo bulb. Valve seat is made of stainless steel, while the needle is of a ball type for perfect seating.

Valve can also be supplied with an auxiliary by-pass feature where off-cycle unloading of the high side is required. Unit weighs 9½ oz.

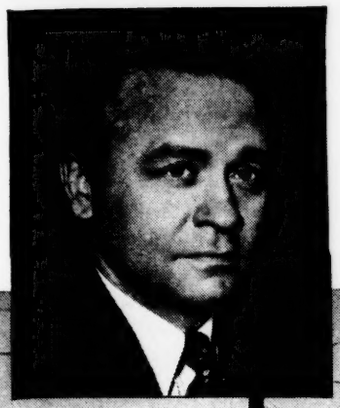
For more information, circle on Inquiry Card, **Item 27**

Safe Boiler Operation

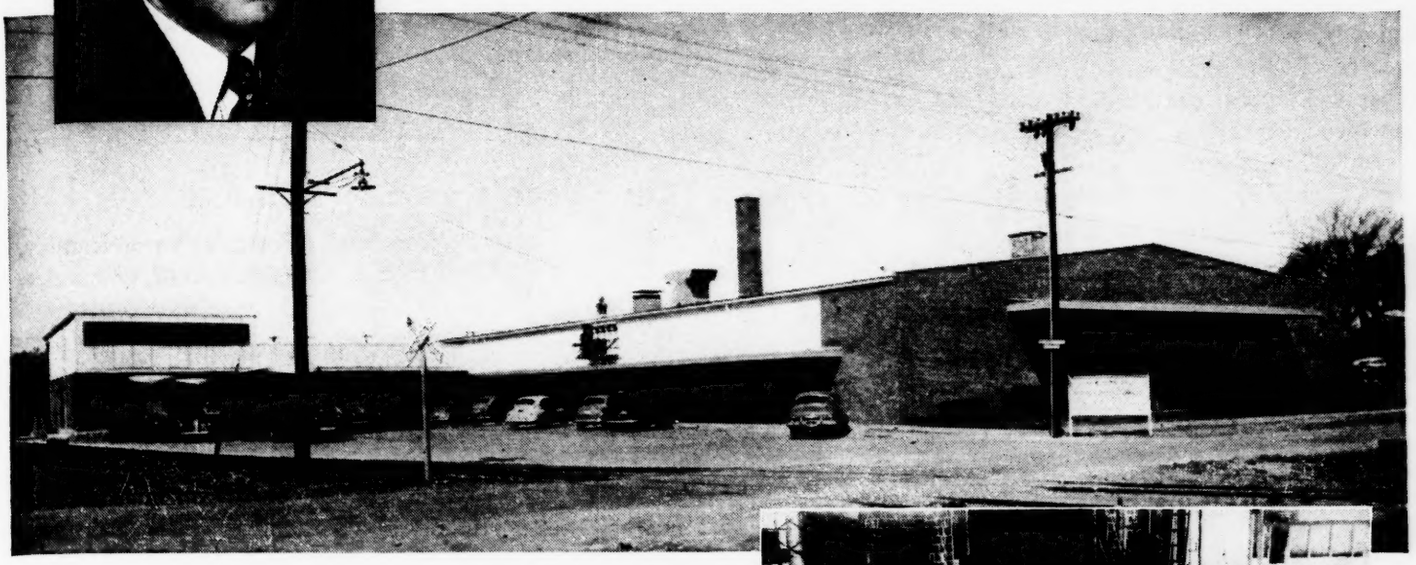
A low water fuel cut-off, McDonnell No. 63, designed to interrupt the flow of current to the oil burner whenever the boiler water line drops below a safe level, is announced by McDonnell & Miller, Inc., Chicago, Ill. In the event of a leak in the hot water heating boiler, or inoperative controls, it stops burner operation.

All operating parts are isolated from the heat of the float chamber. It is designed as a balanced, center-pivoted unit. Body is cast iron, float of heavy copper, and parts of non-corrosive materials. Operating levels are factory set and are non-adjustable. Although unit has an automatic reset, a manual reset can be supplied. It is designed for use with a maximum boiler pressure of 50 lb, and for both hot water and steam boilers. Unit has one-inch pipe taps, top and bottom.

For more information, circle on Inquiry Card, **Item 28**



REVERE DISTRIBUTOR *Takes his own Advice*



...Harry Cooper Supply Co. installs 16,700 feet of Revere Seamless Copper Water Tube for Radiant Panel Heating in its New Office and Showrooms.

For over 15 years Jack Cooper, President of the Harry Cooper Supply Company, Springfield, Mo., has been advising his customers to use Revere Copper Water Tube and other Revere products. He has learned from the many contractors he has sold how it performs . . . its fine record of service. So when he was planning the heating system for his new office and showrooms he picked Revere Copper Water Tube to do the job.

Those who have used Revere Copper Water Tube will tell you that the long lengths of Revere Copper Water Tube make it a real time saver on any radiant panel heating job and that fewer tools are needed to form radiant panel heating coils. Long lengths also mean fewer joints, while light weight simplifies handling. Revere Copper Water Tube cannot rust or rot and has high resistance to corrosion. This means long life. Joints are quickly and easily made with solder fittings, keeping installation costs surprisingly low.

All Revere Copper Water Tube is stamped at regular intervals with the Revere name and the type. Look for these marks, for they are more than identification, they are your assurance of full wall thickness and the close dimensional tolerances so essential for tight soldered joints.

Revere materials are handled by leading distributors in all parts of the country. Your distributor is the man to call if you have a problem and would like to have the Revere Technical Advisory Service help you with it.

REVERE

COPPER AND BRASS INCORPORATED

Founded by Paul Revere in 1801

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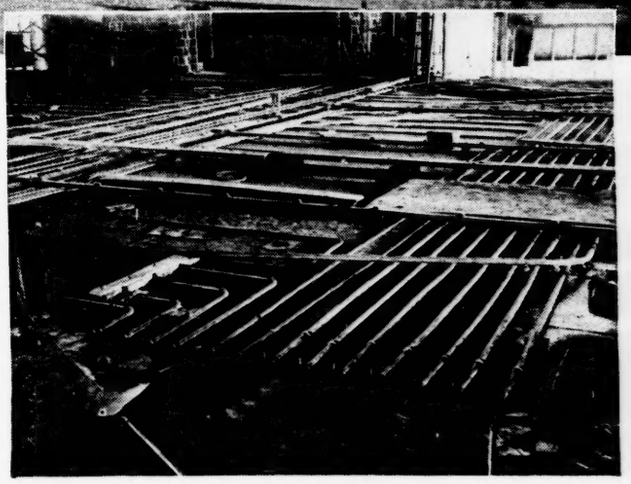
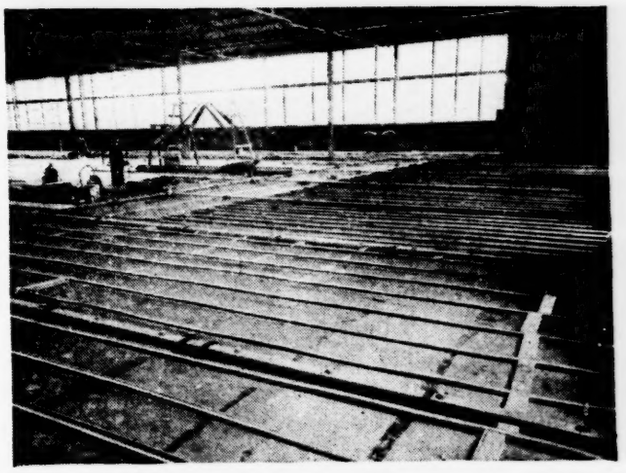
• • •

Mills: Baltimore, Md.; Chicago and Clinton, Ill.; Detroit, Mich.; Los Angeles and Riverside, Calif.; New Bedford, Mass.; Rome, N. Y. Sales Offices in Principal Cities, Distributors Everywhere.



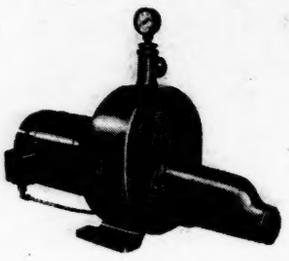
OVER 3 MILES of Revere Seamless Copper Water Tube was used in the radiant panel heating system recently installed in the new office and showrooms of The Harry Cooper Supply Co., Springfield, Mo. Sizes ranged from ¾" to 2".



EVEN, RELIABLE HEAT is assured in all parts of this office by these radiant panel heating coils which were set in a concrete floor slab. Made of Revere Copper Water Tube, coils were easy to bend, needed few fittings, were light to handle. Area of floor panels measures 17,000 sq. ft. Architects, Hawkins & Esterly; Genl. Cont., Chapman & Bramer; Heat. Cont., Bowman & Drussa Plumbing & Heating Co.; Plumbing Cont., Central Plumbing & Heating Co., all of Springfield, Mo.

Pump for 2 Types of Wells

A new series of Delco-Jet convertible pumps is being produced by Delco Appliance Division, General Motors Corp., Rochester, N. Y., available in 1/3 and ½ hp sizes. Pumps are quickly and easily converted from shallow to deep well operation. Shallow well capacities range from 480 to 990 gph; deep well capacities, from 200 to 870 gph.



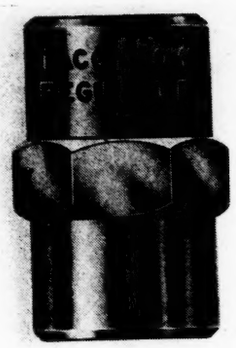
Unit does not have to be located directly over the well itself. Motor and main working parts of the pump are designed for easy removal for repair without disturbing any of the plumbing or pipes.

Pump is operated by a Rigidframe motor with welded stator laminations resulting in quieter operation.

For more information, circle on Inquiry Card, **Item 29**

Flow Regulator in 5 Sizes

To regulate water flow through tankless heaters, the Taco flow regulator has been introduced by Taco Heaters, Inc., Providence 3, R. I. The regulator is of brass and rubber construction with two internal pipe threads and is said to control accurately the flow at any pressure from 15 to 150 lb per sq in.

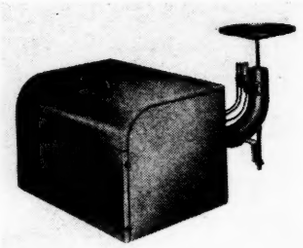


The regulator is made in five sizes as follows: with ½-inch connections, in 3 or 4 gpm capacity; and with ¾-inch connections, in 5, 6 or 8 gpm capacity.

For more information, circle on Inquiry Card, **Item 30**

Gas Burner Made in 2 Sizes

An entirely new series of Delco-Heat gas conversion burners, the GC series, is now being produced by Delco Appliance Division, General Motors Corp., Rochester, N. Y. Burners are available in two sizes with input capacities from 75,000 to 300,000 Btu per hr for natural and mixed gases, and from 75,000 to 225,000 Btu per hr for artificial gas.



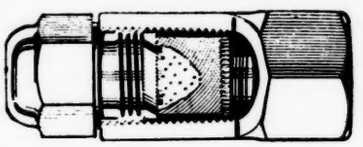
Burner is of the up-shot, single-port type, employing a special stainless steel flame spreader. The adjustable primary air orifice, together with the sliding secondary air shutters, provide positive air-fuel mixture. All controls are contained within a compact, streamlined cabinet which is only 16⅛ inches wide, 11 inches deep and 10½ inches high.

Standard controls include an automatic pressure regulator, automatic main control valve, automatic safety pilot, main shut-off valve, and pilot shut-off valve which includes a gum filter on artificial gas models. An electric pilot lighter is available as an optional extra.

For more information, circle on Inquiry Card, **Item 31**

Oil Filter Can be Cleaned

The Hally nozzle filter, an all metal, cleanable filter, said to eliminate service troubles due to dirty nozzles, has been introduced by the Hally Mfg. Co., Racine, Wis.



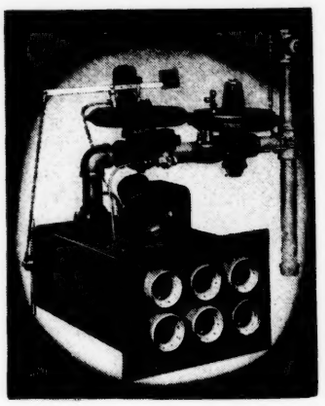
The unit, which screws into the nozzle, replaces the present strainer. It consists of a strainer screen surrounding a stack of 13 rigid grooved metal discs spaced from .00075 inch in the smallest size to .003 inch in the largest size. Discs are clamped tightly together. Each disc contains 12 inlet and outlet grooves. Dirt is separated from the oil and is deposited in the inlet grooves and around the periphery of the filter stack.

Filters are made in several models and sizes for oil ratings from 0.50 to 3 gph.

For more information, circle on Inquiry Card, **Item 32**

Gas Conversion Burner — 7 Sizes

Model 1400 multi-torch gas conversion burners, made by Bryant Heater Division, Affiliated Gas Equipment, Inc., Cleveland, Ohio, are specifically designed for conversion of large size heating boilers and furnaces in apartments, office buildings and commercial establishments to fully automatic gas operation. Burners, shipped complete in one package, permit quick, easy installation and efficient low-cost operation. They are made in seven sizes from 400,000 Btu per hour input to 1,500,000 Btu per hour input.



Units are compact, natural-draft assemblies, designed for ashpit installation. Automatic and smooth in operation, they produce a soft, bushy flame considered ideal for gas firing all types of up-draft boilers and furnaces.

Assemblies consist of a group of injector torches attached to a cast manifold and enclosed in a sheet metal duct.

For more information, circle on Inquiry Card, **Item 33**

WHAT IS THE BIGGEST FACTOR IN SELLING A DUCT JOB ?

We strongly suspect that more sales are won or lost by the contractor's answer to the customer's question of "How much?" than by any other single factor. There may be exceptions, where cost is a secondary consideration with the buyer, but those are not the jobs that build big yearly sales volume.

After weighing the cost, the customer usually wants to know about the sheet material to be used. Will it give long service in ductwork? Is it rigid enough to withstand vibration? Will it look good when it is installed? Will it satisfy all technical requirements?

These questions are all easy to answer when you use Bethlehem Galvanized Steel Sheets. Thousands of heating and air-conditioning jobs extending over a period of many years have proved that Bethlehem Sheets have everything that the customer wants. They give long, trouble-free service — and they are priced well below sheets made from other metals.

BETHLEHEM STEEL COMPANY, BETHLEHEM, PA.

On the Pacific Coast Bethlehem products are sold by Bethlehem Pacific Coast Steel Corporation. *Export Distributor:* Bethlehem Steel Export Corporation

BETHLEHEM STEEL

BETHLEHEM GALVANIZED SHEETS

Duct Sizing is Simplified

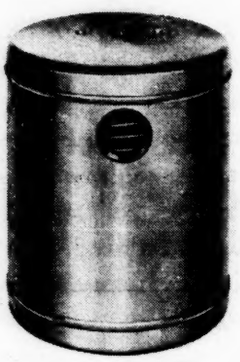
The design steps in sizing ducts have been reduced by the use of the Trane Ductulator made by The Trane Co., La Crosse, Wis. All of the necessary charts are consolidated on a simple, easy-to-use, multiple scaled circular slide rule. An engineer can get all duct sizing answers with a single setting of the Ductulator, which has four colors on four separate scales.

Knowing air volume and velocity, one can find friction loss per 100 ft of duct, diameter of the round duct, and dimensions of a rectangular duct to carry the air load. Any known factor is set to coincide with the known cfm. All other factors may be read with this single setting. Recommended and maximum duct velocities for all general installations appear on the reverse side of the unit. Ductulator is supplied at cost.

For more information, circle on Inquiry Card, **Item 34**

Tankless Water Heaters

Two new tankless water heaters have been added to the line of Bell & Gossett Co., Morton Grove, Ill. Heaters can be furnished as a standard unit without the jacket or as a deluxe unit (illustrated) with a well-insulated metal jacket. They have a cast iron shell and a spiral copper tube bundle.



The three sizes which are available have capacities of 3½, 4, and 5 gpm, when water is heated from 40 to 140F and the boiler water or heating medium is at 180F. This capacity is increased to 5, 6, and 7½ gpm, respectively, when the boiler water is at 200F.

For more information, circle on Inquiry Card, **Item 35**

Rustless Glass-Fiber Pipe

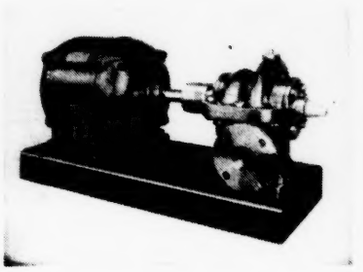
Development of a glass-fiber tube and pipe material, designed as a replacement for steel and other critical war metals in many commercial applications, is announced by United States Plywood Corp., New York, N. Y. Known as Glasweld, this product has the strength of steel and is rust and corrosion proof. It is available as piping in the oil and chemical processing industry and as tubing for the building, electrical and allied fields.

The product is a laminated tubing in which glass fibers, in the form of cloth, mat or tape, are bonded with resins to develop a rugged tube impervious to heat, chemical action and sledgehammer blows. It has good resistance, it is claimed, to many dilute and strong acids.

For more information, circle on Inquiry Card, **Item 36**

Service Pump Cuts Repair

Designed so that all parts subject to wear can be removed in one rotor assembly, the De Laval type GS service pump made by De Laval Steam Turbine Co., Trenton, N. J., cuts maintenance costs. It comes in three sizes for capacities to 400 gpm and heads to 230 ft.



When maintenance is necessary, the top cover and end plate bolts are removed, the worn assembly lifted out, and a new one installed. Through an exchange plan, new rotors are available from stock.

Shaft is of stainless steel, and the pump case and cover of cast iron. Pump impeller is bronze, shrunk and keyed to the shaft. The two end covers are constructed to serve as a guide for the liquid to enter the impeller smoothly; enclose and support the mechanical seal; serve as housing for bearing. Simple mechanical seals replace the stuffing box.

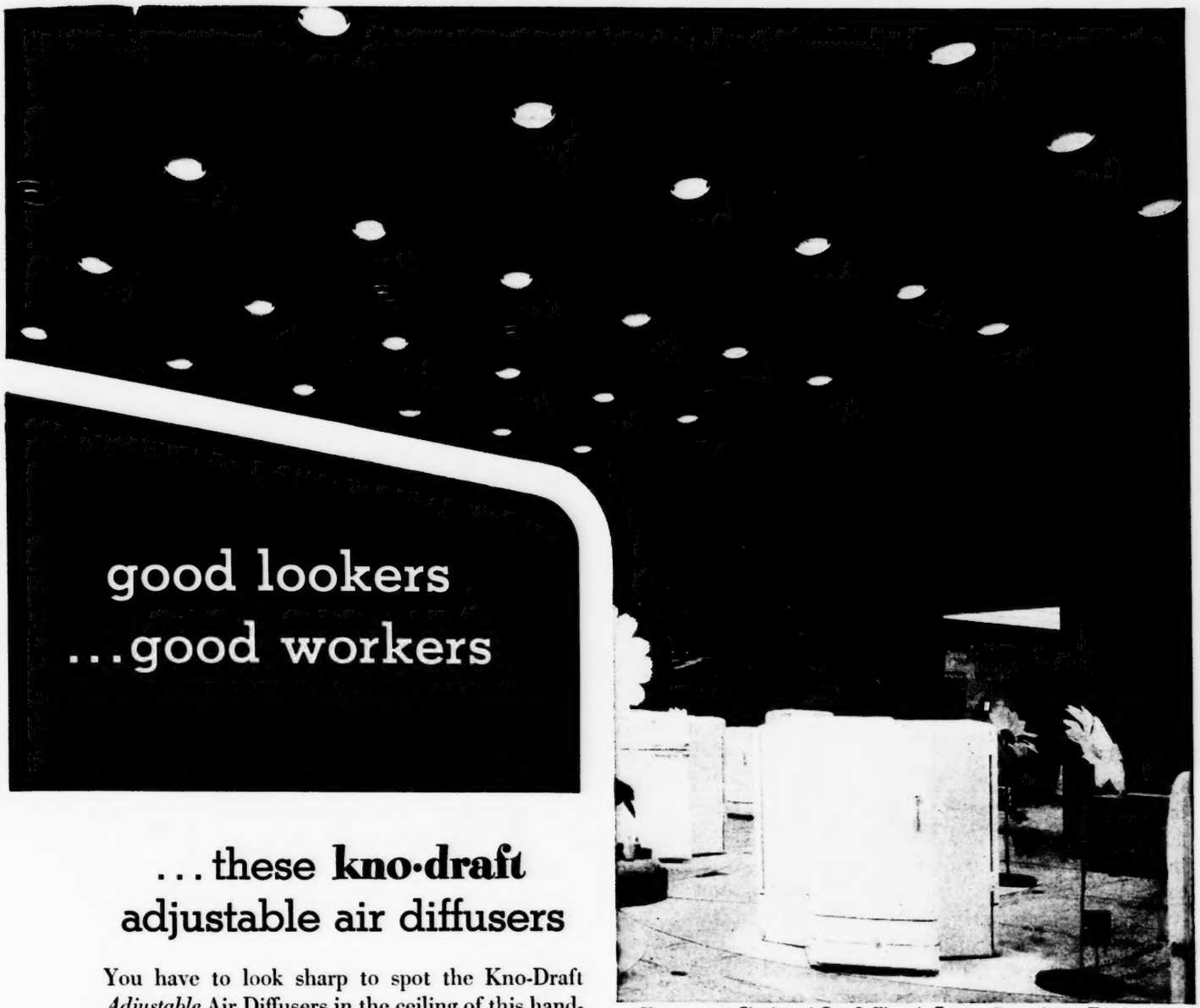
For more information, circle on Inquiry Card, **Item 37**

Easily Serviced Oil Burner

The Model 400S oil burner has been introduced by the newly formed United States Burner Division of The Carlin Co., Wethersfield, Conn. It incorporates the Shell high temperature combustion head and electromagnetic fuel control that combine toward greater efficiency and economy of operation.

One of the features is the Verni-flame control that provides vernier flame adjustment with an ordinary hex wrench at the rear of the burner. There is nothing to take apart. Servicemen can completely eliminate feathering and obtain a perfect cone flame simply by turning the adjustment screw. The burner is also easy to service. The electrode assembly can be removed or replaced in less than two minutes. Bus bars replace wires. Motorfan unit can be quickly removed by unscrewing just two bolts. The burner is available in two capacities. At 1,725 rpm, Model 400S-2 is rated at 0.75 to 1.75 gph, while Model 400S-3 is rated at 1.75 to 3 gph.

For more information, circle on Inquiry Card, **Item 38**



good lookers ...good workers

...these **kno·draft** adjustable air diffusers

You have to look sharp to spot the Kno-Draft *Adjustable* Air Diffusers in the ceiling of this handsome public utility showroom . . . so well do they blend in with the architect's design for lighting.

Yet these good lookers are good workers, too. None better, in fact. Kno-Draft *Adjustable* Air Diffusers circulate the air gently and evenly . . . without drafts to bother persons on the floor . . . without differences in temperature between one spot and another.

This pattern of air flow—both its volume and direction—can be adjusted *after* installation. A few turns of a screw driver thus replace lengthy slide rule calculations.

Showroom: Cincinnati Gas & Electric Co. • **Consulting Engineers:** Wm. E. Bodenstein and W. W. Shuster • **Contractor:** The Cincinnati Air Conditioning Company

You can paint Kno-Draft *Adjustable* Air Diffusers or leave them in their natural aluminum . . . whichever goes best with the established decorative theme. You can get them in suitable types and sizes for every job.

KNO-DRAFT DATA BOOK: Complete specifications, engineering and installation data on Kno-Draft *Adjustable* Air Diffusers. To get your copy, simply fill in and mail the coupon. No obligation, of course.

TRADE MARK "KNO-DRAFT" REG. U. S. PAT. OFF.

W. B. CONNOR ENGINEERING CORP.
Danbury, Connecticut
Air Diffusion • Air Purification • Air Recovery
In Canada: Douglas Engineering Co., Ltd., 190 Murray Street, Montreal 3, P. Q.

W. B. CONNOR ENGINEERING CORP.
Dept. J-100, Danbury, Connecticut

Please send me without obligation, my copy of the Kno-Draft Air Diffuser Data Book.

Name.....

Position.....

Company.....

Street.....

City..... **Zone**.....**State**.....

NEWS OF THE MONTH

GAS UTILITIES

expect to add more than three million house heating customers by 1953

The gas utilities expect to add more than 3,000,000 new house heating customers in the next three heating seasons to the 7,217,000 customers who now heat their residences with gas, H. Carl Wolf, managing director of the American Gas Association, announced today. Canadian gas companies serving 225,000 retail customers hope to add about 13,000 gas house heating customers in the three-year period.

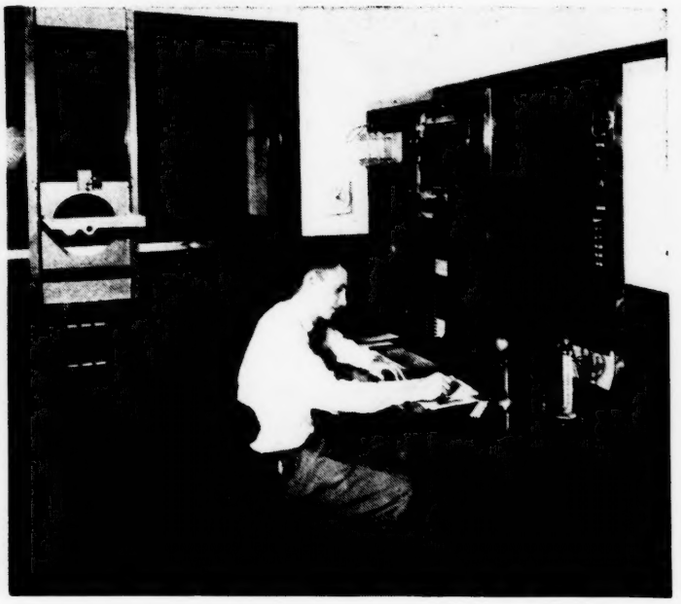
• **MARKET.**—The A.G.A. Bureau of Statistics has just completed a survey which indicated that the gas utilities in America will serve 1,150,000 more house heating customers during the coming 1950-1951 heating season than it did last winter. This means a total of 9,400,000 residential gas customers, or more than 40% of all dwelling units being supplied with utility gas service, are expected to heat their homes with gas during the coming winter.

Prior to the accelerated military preparedness program, with its impending steel allocations, the gas industry expected to add 950,000 house heating customers in the 1951-1952 heating season and another 900,000 house heating customers in the 1952-1953 heating season. The Canadian companies surveyed expect to add 5,000 customers, 4,000 customers and 4,000 customers, respectively, in the three seasons.

• **SURVEY.**—These estimates were based on replies of 295 individual gas utilities in the United States serving about 19,750,000 gas customers, or about 90% of the nation's total. In addition, six Canadian gas companies submitted their house heating plans.

The reporting gas companies indicated that approximately half of the anticipated gas house heating additions were to be installed in new dwelling units with the other half representing conversions from other fuels.

• **PIPELINES.** — During the past twelve months several new pipelines, or extensions of previously existing lines were completed. This has enabled many natural gas distributing utilities to lift, either wholly or in substantial part, existing restrictions on additional gas house heating installations. The availability of natural gas in new areas and continued construction of pipelines can be expected to accelerate further new gas heating installations contingent upon movement of materials as affected by the war situation.



AIR DIFFUSION PATTERNS will be studied in a new laboratory equipped for testing of full scale mock-ups of rooms at Pyle-National Company's Chicago plant. Heat losses or gains are imposed by varying the temperature of the air surrounding a room within a room. Air at closely controlled temperatures is accurately metered into the mock-ups through diffusers for tests of the effectiveness of distribution. One large test space houses a room with a movable floor wherein air currents and temperatures can be determined remotely and the effect of various ceiling heights from 8 to 16 ft studied. Another test space is equipped for models of railroad cars or other types of enclosure. A two-level structure between the two test spaces houses refrigerating equipment, blowers, etc., on the lower level and a control station on the upper. The desired test conditions are set from the control room. Large windows into each of the test rooms allow visual observation. Remote reading air velocity and temperature instruments are located on the control panel. In the test space, the mixing chamber of a small smoke generator and exploring thermocouples are located on a slider which can be electrically moved along a transverse bar. The bar also has fore and aft motion along a supporting frame which in turn can be raised or lowered. Thus, the operation of three motors allows the slider to be pinpointed anywhere.

GAS HOUSEHEATING SURVEY, 1950-1953

Census Region	Customers of Companies Included in Survey as of April 1, 1950		Anticipated Additional Househeating Customers		
	Total	House-heating	'50-'51 Season	'51-'52 Season	'52-'53 Season
1. New England	1,293,266	56,300	10,576[1]	12,686[1]	17,823[1]
2. Middle Atlantic	5,867,705	616,016	118,397	114,249	104,984
3. East North Central	4,754,065	1,097,091	323,755	206,531	181,874
4. West North Central	1,429,275	634,113	137,024	123,413	111,741
5. South Atlantic	1,290,882	396,746	88,928	87,203	77,082
6. E. S. Central	585,521	342,885	46,398	46,872	35,831
7. W. S. Central	1,588,065	1,552,006	126,368	124,066	123,372
8. Mountain	366,178	289,683	27,766	24,748	18,221[2]
9. Pacific	2,572,601	2,232,384	133,930	112,915[3]	103,524[3]
U. S. Total	19,747,558	7,217,224	1,013,142	852,683	774,452

[1]Vermont not included [2]Utah not included [3]Oregon not included

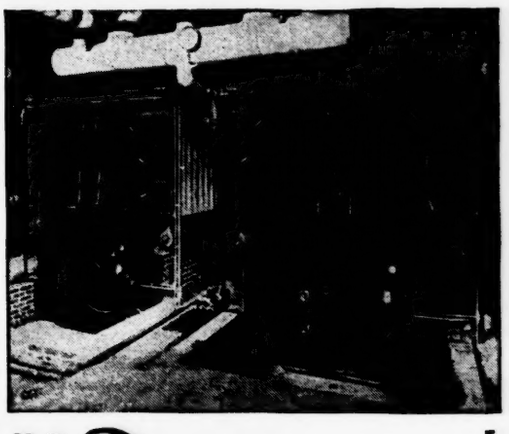
Hold it!

you don't need a Tall Chimney with an H. B. SMITH BOILER

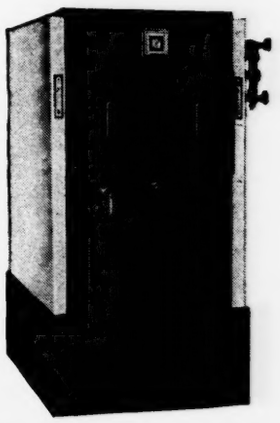
To keep the lines of that modern functional school within the proper architectural proportions — and to give your client truly outstanding heating plant performance — we suggest an H. B. Smith boiler.

The vertical water tube design of H. B. Smith boilers provides minimum draft resistance. Hot gases rise naturally between the boiler sections before entering the side flues. This positive gas travel is an advantage not found in boilers of the return-flue or long-fire-tube type, in which draft loss through the boiler is much greater. Tall chimneys or complicated induced-draft controls are, therefore, not necessary with H. B. Smith boilers.

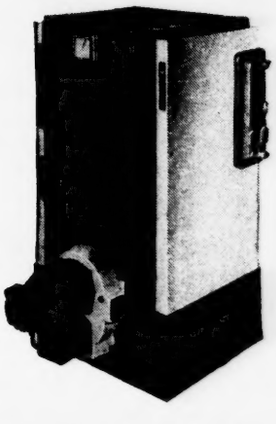
For added assurance of economy of operation and long-life performance in schools or other buildings, consider the many, many installations in older schools where H. B. Smith Cast Iron Boilers have been giving top performance for 25, 35, even 50 years! You can recommend an H. B. Smith boiler for the life of the building!



A modern school equipped with No. 60 Smith Boilers.



SMITH-MILLS "20" BOILER for Hand Firing. These smaller H. B. Smith Boilers have the same vertical flue travel, hence the same low-draft requirements as do the famous big "MILLS" Boilers.



SMITH-MILLS "1100" BOILER-BURNER UNIT. A complete heating plant for the average home — equipped with oil burner and tankless heater. Extended type jacket available.

You may be interested to know that H. B. Smith manufactures the most complete line of modern cast iron boilers in the world. Let us help you satisfy your customers with convincing data and case histories.

H. B. Smith

CAST IRON BOILERS

THE H. B. SMITH CO. INC., WESTFIELD, MASS.

Most complete line in the world of cast iron boilers for heating

HEATING SYSTEMS

for new homes surveyed by shelter magazine.

A survey of the types of heating systems being used in new homes was recently completed by the magazine *Better Homes & Gardens.* The survey was made on a national basis with questionnaires going to families building new homes.

DISTRIBUTION OF HEATING SYSTEMS BY TYPE

Region	Space Heater	Warm Air With Blower	Warm Air Without Blower	Radiators (Steam or Hot Water)	Radiant[1] Heat	Other
New England	1%	33%	6%	57%	2%	1%
Middle Atlantic	—	42	9	43	5	1
South Atlantic	16	49	6	16	4	9
East North Central	2	63	11	7	13	4
East South Central	2	39	19	6	14	20
West North Central	2	74	10	7	4	3
West South Central	13	25	12	3	20	27
Mountain and Pacific	11	44	16	2	15	12
Total U. S.	6%	49%	11%	16%	10%	8%

[1]Includes radiant heat in floors, ceiling, or baseboards.
Percentages based on number reporting type of heating.

As shown in the table, the incidence of radiator heating in New England and Middle Atlantic states is higher than the national average. In New England, heating with radiators (57%) and radiant heat (2%) make 59% of the new homes heated with boilers as against 39% with warm air furnaces (33% warm air with blowers and 6% warm air without blowers).

In the Middle Atlantic states 48% of the new homes use boilers (43% radiator heat and 5% radiant heat), while 51% use warm air furnaces (42% warm air with blowers and 9% warm air without blowers).

METERED WATER USE

determined for various families in HHFA study of sewage disposal.

In a study of individual sewage disposal systems, the Housing Technology Branch of the Housing and Home Finance Agency surveyed 170 residences for water consumption per capita in winter and summer. Purpose of the survey was to serve as a guide to the amount of sewage being discharged. The survey was part of a broader study of septic tanks in field applications which is covered in Technical Paper No. 14, May, 1950, prepared by Laurence Shuman and Fred W. McGhan of the Housing Technology Branch.

TABLE 1—METERED WATER USE—GALLONS PER CAPITA PER DAY.

Persons Resident	No. of Reports	Summer				Winter			
		Mean	Max.	Median	Min.	Mean	Max.	Median	Min.
1	1	58				50			
2	26	78	226	62	16	53	100	48	8
3	45	50	149	44	14	41	106	36	12
4	58	41	104	39	8	34	85	30	12
5	24	42	170	41	15	35	150	34	13
6	9	36	109	30	20	30	77	25	15
7	5	37	69	35	12	28	55	25	10
8	1	32				22			
9	1	23				23			
	170	48.7	226		8	38.5	150		8

A summary of the water-use survey is presented in Table 1. It is interesting to note that for the 170 installations for which summer and winter figures were available, the daily summer usage, averaging 48.7 gallons per capita, closely approached the 50 gallons per capita sewage flow figure generally used in household septic tank design. Data show that the average water consumption per person decreases with increasing size of family.

COLD BELT RADIANT HEATING will be tested in a made-to-order laboratory at Bangor, Maine. The "laboratory" is a 15-room, one-story annex to Bangor's Fairmont School. The annex is radiant heated throughout, including six classrooms. The new structure was added to a multi-story brick building which is steam heated. The annex includes hundreds of square feet of glass window area, intended to provide the best possible natural lighting. The generous use of glass also presented a heating problem which is normallly difficult in these cold weather areas. Eaton Tarbell & Associates were architects and engineers on this project. Operation of the heating system has been so pleasing to date that the same firm is now designing a radiant heating system for the subprimary and first and second grades of another school for the City of Bangor.

• Acme Industries, Inc., has announced the election of Guy L. Baldwin as vice president of the company.

• Stephen F. Keating, director of military contracts for Minneapolis-Honeywell Regulator Co., has been elected an assistant secretary of the company. Mr. Keating joined Honeywell in 1948 as executive assistant in the Aeronautical division. Previously he had been a member of the law firm of Otis, Faricy and Burger, St. Paul, Minnesota.

• Emil Herman Erickson, 74, vice president and director of The Trane Co., La Crosse, Wis., died early Tuesday, Sept. 12, after a long illness. As general superintendent, Mr. Erickson actively directed Trane's manufacturing operations until World War II. Joining Trane in 1918, he was largely responsible for the company's entering the manufacturing field. Miles L. Erickson and Thomas Hancock have been elected vice presidents of The Trane Co. Mr. Erickson has been employed by the company since 1935. He is the son of the late E. H. Erickson, one of the founders of The Trane Company. Tom Hancock joined the organization in 1934 as an assistant in the advertising department, after which he was transferred to a sales position. He is now in full charge of all company sales.

THRESHOLD LIMIT VALUES

of air contaminants adopted at annual meeting of governmental industrial hygienists.

Threshold limit values (formerly called maximum allowable concentrations) of gases, vapors, dusts, fumes, and radiation were revised and adopted at the April, 1950, meeting of the American Conference of Governmental Industrial Hygienists in Chicago. While most values set up at the 1947 meeting remained unchanged in the new tabulation, substantial changes percentagewise were made in values for about a dozen substances.

• **CHANGES.**—Some of the values formerly established as maximum allowable concentrations were increased while others were decreased. Values increased were for: *n*—Butanol (50 to 100 ppm), 2—butanone (200 to 250 ppm), isopropyl ether (400 to 500 ppm), antimony (0.1 to 0.5 mg per cu M), arsenic 0.1 to 0.5 mg per cu M), and sulfuric acid 0.5 to 1.0 mg per cu M).

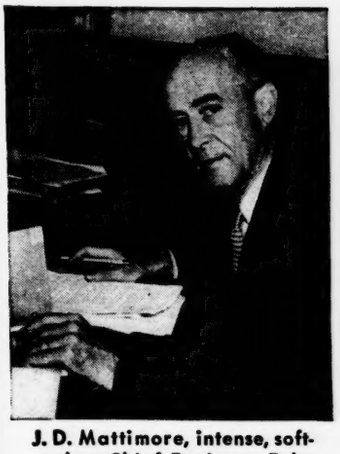
Decreased values were for: Benzene (50 to 35 ppm), chlorine (2 to 1 ppm), formaldehyde (10 to 5 ppm), hydrogen chloride (10 to 5 ppm), hydrogen cyanide 20 to 10 ppm), hydrogen selenide (0.1 to 0.05 ppm), and soapstone below 5% free SiO_2 (50 to 20 million particles per cubic foot of air).

Threshold values for gamma and X-ray radiation were changed from 0.1 roentgen per day to 0.3 roentgen per week.

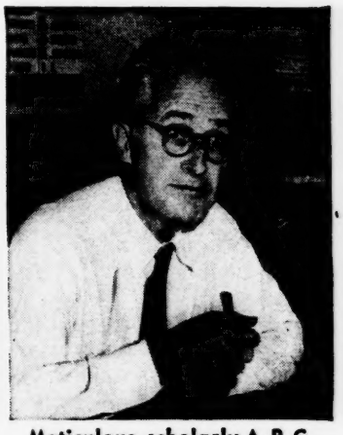
The complete table of values is shown on page 126.

• **COMMITTEE.**—The committee set up to examine threshold limit values and submitting the approved report consisted of L. T. Fairhall, chairman; C. E. Couchman, L. F. Garber, F. R. Ingram, and J. Shilen.

These men are working for you



J. D. Mattimore, intense, soft-spoken Chief Engineer, Tube Turns' Product Engineering and Research Division.



Meticulous, scholarly A. R. C. Markl, Chief Research Engineer, Product Engineering and Research Division.

Thorough, analytical Arthur McCutchan, Chief Product Engineer, Product Engineering and Research Division.

T. D. Bridge, quiet, studious Product Engineer, Product Engineering and Research Division.

LIKE MOST U. S. MANUFACTURERS, Tube Turns, Inc. is sincerely determined to bring its customers the finest products possible. Fulfillment of that purpose is largely the responsibility of its research and product engineers.

No general practitioners, these men are recognized piping authorities, possessing highly specialized knowledge and training in the field of piping engineering. Together, they represent a total of more than 100 years experience in the design and application of piping, fittings, and flanges for power, oil, chemical, marine, and general industrial service.

Aided and abetted by a corps of trained technicians, their efforts are consistently directed toward one never-changing goal: *better welding fittings and flanges to serve you better!*

Are you on the mailing list?

If you are concerned with designing or erecting piping systems, you'll want to receive new "Piping Engineering" bulletins. Just write us on your business letterhead, giving name, title, address.

"Be Sure You See The Double tt"

TUBE TURNS, INC.

LOUISVILLE 1, KENTUCKY

THRESHOLD LIMIT (M.A.C.) VALUES OF AIR CONTAMINANTS

The following threshold limit values, formerly designated maximum allowable concentration values (M.A.C.), were those adopted by the American Conference of Governmental Industrial Hygienists at its 1950 meeting. The table is reproduced by permission of the Conference.

GASES AND VAPORS

Substance	*M.A.C. (ppm*)*
Acetaldehyde	200
Acetic acids	10
Acetic anhydride	5
Acetone	500
Acrolein	0.5
Acrylonitrile	20
Ammonia	100
Amyl acetate	200
iso-Amyl alcohol	100
Aniline	5
Arsine	0.05
Benzene (benzol)	35
Bromine	1
1,3-Butadiene	1,000
n-Butanol	100
2-Butanone	250
n-Butyl acetate	200
Butyl "cellosolve"	200
Carbon dioxide	5,000
Carbon disulfide	20
Carbon monoxide	100
Carbon tetrachloride	50
"Cellosolve"	200
"Cellosolve" acetate	100
Chlorine	1
2-Chlorobutadiene	25
Chloroform	100
1-Chloro-1-nitropropane	20
Cyclohexane	400
Cyclohexanol	100
Cyclohexanone	100
Cyclohexene	400
Cyclopropane (propene)	400
0-Dichlorobenzene	50
Dichlorodifluoromethane	1,000
1,1-Dichloroethane	100
1,2-Dichloroethane (ethylene dichloride)	75
1,2-Dichloroethylene	200
Dichloroethyl ether	15
Dichloromethane	500
Dichloromonofluoromethane	1,000
1,1-Dichloro-1-nitroethane	10
Dichlorotetrafluoroethane	1,000
Dimethylaniline	5
Dimethylsulfate	1
Dioxane	100
Ethyl acetate	400
Ethyl alcohol	1,000
Ethyl benzene	200
Ethyl bromide	200
Ethyl chloride	1,000
Ethyl ether	400
Ethyl formate	100
Ethyl silicate	100
Ethylene chlorohydrin	5
Ethylene oxide	100
Formaldehyde	5
Gasoline	500
Heptane	500
Hexane	500
Hydrogen chloride	5
Hydrogen cyanide	10
Hydrogen fluoride	3
Hydrogen selenide	0.05
Hydrogen sulfide	20
Isophorone	25
Mesityl oxide	50
Methanol	200
Methyl acetate	200
Methyl bromide	20
Methyl butanone	100
Methyl "cellosolve"	25
Methyl "cellosolve" acetate	25
Methyl chloride	100
Methylcyclohexane	500
Methylcyclohexanol	100
Methylcyclohexanone	100
Methyl formate	100
Methyl iso-butyl ketone	100
Monochlorobenzene	75
Monofluorotrichloromethane	1,000
Mononitrotoluene	5
Naphtha (coal tar)	200
Naphtha (petroleum)	500
Nickel carbonyl	1
Nitrobenzene	1
Nitroethane	100
Nitrogen oxides (other than nitrous oxide)	25
Nitroglycerine	0.5
Nitromethane	100
2-Nitropropane	50
Octane	500
Ozone	1
Pentane	1,000
Pentanone (methyl propanone)	200
Phosgene	1
Phosphine	0.05
Phosphorus trichloride	0.5
iso-Propanol	400
Propyl acetate	200
iso-Propyl ether	500
Stibine	0.1
Stoddard solvent	500
Styrene monomer	200
Sulfur chloride	1
Sulfur dioxide	10
1,1,2,2-Tetrachloroethane	5
Tetrachloroethylene	100
Toluene	200
Toluidine	5
Trichloroethylene	150
Turpentine	100
Vinyl chloride (chloroethane)	500
Xylene	200

TOXIC DUSTS, FUMES, AND MISTS

Substance	*Mg/M³***
Antimony	0.5
Arsenic	0.5
Barium	0.5
Cadmium	0.1
Chlorodiphenyl	1
Chromic acid and chromates (as CrO_3)	0.1
Cyanide as CN	5
1,2-Dichloropropane (propylene dichloride) (ppm)	75
Dinitrotoluene	1.5
Fluorides	2.5
Iodine	0.1
Iron oxide fume	15
Lead	0.15
Magnesium oxide fume	15
Manganese	6
Mercury	0.1
Pentachloronaphthalene	0.5
Pentachlorophenol	0.5
Phosphorus (yellow)	0.1
Phosphorus pentachloride	1
Phosphorus pentasulfide	1
Selenium, compounds as selenium	0.1
Sulfuric acid	1.0
Tellurium	0.1
Tetryl	1.5
Trichloronaphthalene	5
Trinitrotoluene	1.5
Zinc oxide fumes	15

MINERAL DUSTS

Substance	*M.P.P.C.F.**
Alundum	50
Asbestos	5
Carborundum	50
Dust (nuisance, no free silica)	50
Mica (below 5% free silica)	50
Portland cement	50
Silica—high (above 50% free SiO_2)	5
Silica—medium (5 to 50% free SiO_2)	20
Silica—low (below 5% free SiO_2)	50
Slate (below 5% free SiO_2)	50
Soapstone (below 5% free SiO_2)	20
Talc	20
Total dust (below 5% free SiO_2)	50

RADIATIONS

Material or radiation	*Radiant energy*
Gamma radiation	0.3 roentgen per week.
Radon	10^{-8} curies per cubic meter.
Thoron	10^{-8} curies per cubic meter.
X-ray	0.3 roentgen per week.

*ppm = parts per million.

*Mg/M³=milligrams per cubic meter.

*M.P.P.C.F. = million particles per cubic foot of air, standard light field count.

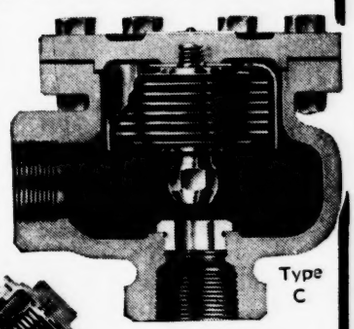
Large Processor Achieves

MORE UNIFORM HEATING CYCLES

with Nicholson Steam Traps

The positive action of Nicholson traps was recently used successfully by a chemical maker to smooth out the heating cycles of cookers at 50 to 60 minutes, against previous periods up to 105 minutes. Steam flow was upped to 3000 lbs. an hour against 2000 for the previous mechanical type.

5 TYPES FOR EVERY APPLICATION, process, heat, power. Sizes ¼" to 2"; pressure to 225 lbs. Send for Bulletin 450.



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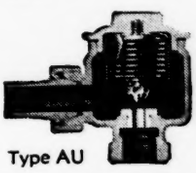

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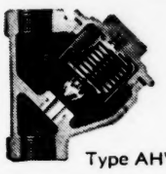

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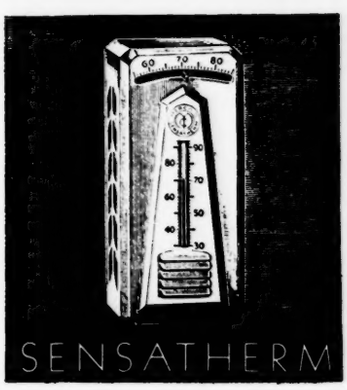
BULLETIN 450 or see Sweet's

W. H. NICHOLSON & CO. 199 OREGON STREET WILKES-BARRE, PA.

Sales and Engineering Offices in 53 Principal Cities

If they are,



MERCOID CONTROLS

You can always count on their fine engineering to give very dependable service for a long time. They are built to last.

Complete catalog sent upon request

THE MERCOID CORPORATION
4201 BELMONT AVE. CHICAGO 41, ILL.

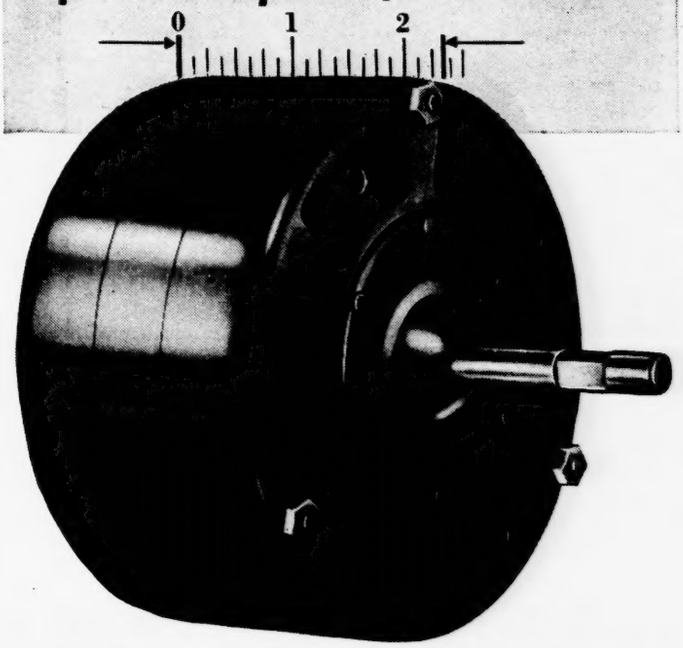
NOW AVAILABLE

in "SPACE-SAVER" and REVERSIBLE MODELS

FASCO

6-POLE SHADED POLE MOTORS

1/30 THRU 1/8 HP, 1000 RPM



- **NEW! FASCO "SPACE-SAVER"... FLAT, COMPACT, POWERFUL**
- **NEW! FASCO REVERSIBLE... AVAILABLE IN ALL 6-POLE MODELS**

★ Quiet, uniform performance at all speeds

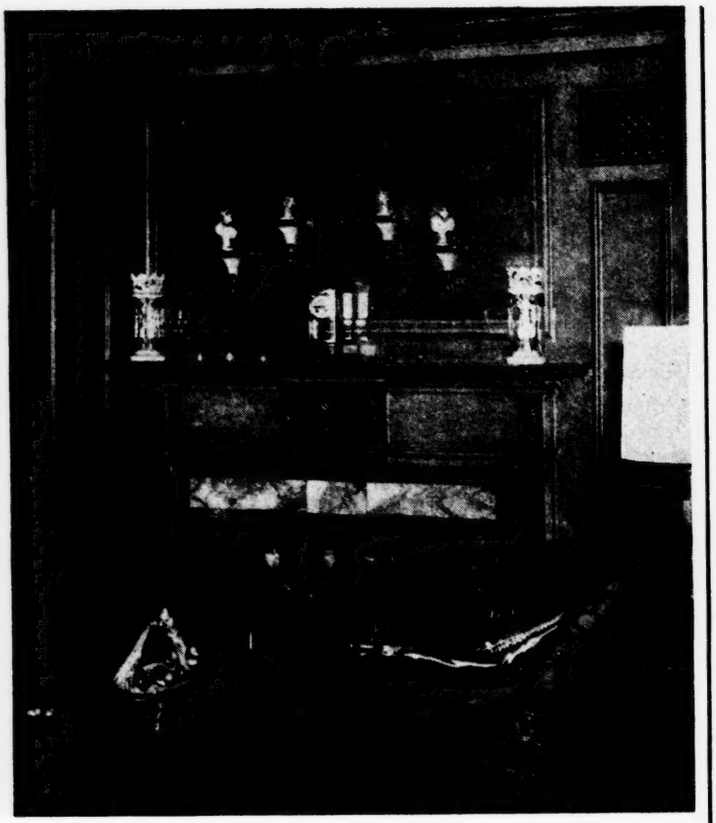
★ Speed reductions to 500 RPM

★ Higher efficiency . . . long life

★ Mounts to meet every need . . . interchangeable with other motors

WRITE today for full performance data, dimensions, and applications to FASCO Industries, Inc., 286 Toppin St., Rochester 2, New York.

FASCO
INDUSTRIES, INC.
ROCHESTER 2, NEW YORK



Perfect Air Control with Ornamental Outlets

For air circulating systems in residences the dual-unit Hendrick Bulator is ideal. It provides complete control of air throw and spread, with a decorative effect that actually adds to the attractiveness of the room.

The installation shown is in the home of the president of a prominent corporation. Although the deflecting vane grilles of the Bulator are directly behind the Gothic Cloverleaf design of the ornamental grilles, only the latter are visible from the room.

For Bulator dual-unit installations a wide variety of ornamental grille designs are available, so one can readily be selected to harmonize with the decorative scheme of any room.

Write for full details on this practical dual-unit combination.

See our catalog in SWEET'S

Perforated Metals
Perforated Metal Screens
Architectural Grilles
Mitco Open Steel Flooring,
"Shur-Site" Treads and
Armorgrids

HENDRICK
Manufacturing Company
42 DUNDAFF STREET, CARBONDALE, PA.
Sales Offices in Principal Cities

ENGINEERS OF ASHVE

to probe physiological aspects of air conditioning at annual meeting.

The physiological aspects of air conditioning will be discussed in an important group of papers at the 57th Annual Meeting of The American Society of Heating and Ventilating Engineers in Philadelphia, Jan. 22-25, 1951. It is anticipated that the papers will cover physiological principles, requirements for comfort, air conditioning for the treatment and prevention of disease, and industrial hygiene.

Other topics proposed for discussion at the meeting's five technical sessions will include smoke measurement, solar radiation, air flow and its measurement, and heat pump performance. Titles and authors of papers will be announced at a later date. Meeting headquarters will be at the Bellevue-Stratford Hotel. The Philadelphia Chapter of the Society will act as host, with A. J. Nesbitt, president of John J. Nesbitt, Inc., of Philadelphia, as general chairman of the committee on arrangements.

Merrill F. Blankin, president of the Blankin Equipment Corp., of Philadelphia, and a past president of the Society, has been named honorary chairman and F. H. Buzzard, chief engineer of Charles S. Leopold, also a Quaker City firm, will serve as vice chairman.

• **LARGEST DISPLAY.**—The Tenth International Heating and Ventilating Exposition, also known as the Air Conditioning Exposition, will be held in conjunction with the ASHVE meeting at the Commercial Museum, Jan. 22-26, 1951. According to C. F. Roth, manager of the exposition, it will be the largest display of heating, ventilating and air conditioning equipment ever to be housed in one place. Lester T. Avery, president of ASHVE, is chairman of the advisory committee for the exposition.

The Philadelphia Chapter has been host to the Society for three previous annual meetings, in 1921, 1930 and 1942.

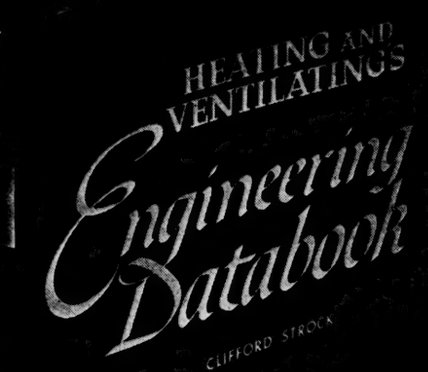
Serving as chairmen of the various committees of the committee on arrangements are: banquet, M. E. Barnard; entertainment, E. K. Wagner; exposition, C. F. Dietz; finance, A. C. Caldwell; ladies, E. H. Dafter; publicity, W. P. Culbert; reception, J. O. Kirkbride; sessions, John Everetts, Jr., and transportation, Isadore Director.

WARM AIR MEETING

will feature talks on industry conditions.

Plans for the 37th Annual Convention of the National Warm Air Heating and Air Conditioning Association are nearing completion, according to Clarence S. Franke, president of the association and Irving E. Seith, chairman of the program committee. The convention is scheduled for Tuesday and Wednesday, December 5th and 6th, at Hotel Statler in St. Louis.

• **PROGRAM.**—One of the main speakers is Dr. Arthur H. Compton, Chancellor of Washington University, St. Louis. Dr. Compton is one of the world's top physicists and a forceful speaker. He has won a Nobel



HEATING AND VENTILATING'S Engineering Databook

HEATING AND VENTILATING'S Engineering Databook is an authoritative reference work, providing data on heating, ventilating and air conditioning for design, installation and operating engineers. Its 576 pages contain 317 essential tables, 193 charts and illustrations, and the 8 1/2" x 11" page size provides space for large-scale, easy-to-read charts, which eliminate or greatly simplify many everyday estimating and design problems. Subjects are grouped under the following eleven main divisions, which briefly illustrate the great variety of practical information in this valuable reference work:

- Heat and Moisture in Air
- Fuels and Combustion
- Piping – Sizing, Standards, etc.
- Air Handling and Ventilation
- Heating and Heat Transmission
- Cooling and Air Conditioning
- Motors – V-Belt Drives
- What the Contractor Should Know About Contracts
- Mathematical Tables
- Drafting Room Standards
- Terminology of the Industry

The Engineering Databook is filled with fundamental information, but its overall scope is considerably more than the covering of commonly used data. In addition, the most up-to-date information available is presented on the important subjects shown in the following list:

- The Heat Pump
- Electrothermal Storage
- Snow Melting
- Heating with Diesel Engines
- Attic Ventilation
- Heating Water by the Sun
- Radiant (Panel) Heating
- Basementless Buildings
- Axial Flow Fans
- Estimating for Air Conditioning Units
- Toxicity
- Law of Contracts

You will find use every day for the comprehensive, down-to-earth information in your Engineering Databook. Place your order now for this authoritative book. The coupon below—with refund offer in the United States – is for your convenience.

576 Pages

510 Tables, Charts and Illustrations

THE INDUSTRIAL PRESS, 148 Lafayette Street, New York 13, N. Y.

I enclose check, postal note or money order for $....., (include 85c Canadian or foreign postage if necessary). Please send me a copy of HEATING AND VENTILATING'S ENGINEERING DATABOOK. I have the right to return the book within 5 days and money will be refunded.

Name.....

Home Address.....

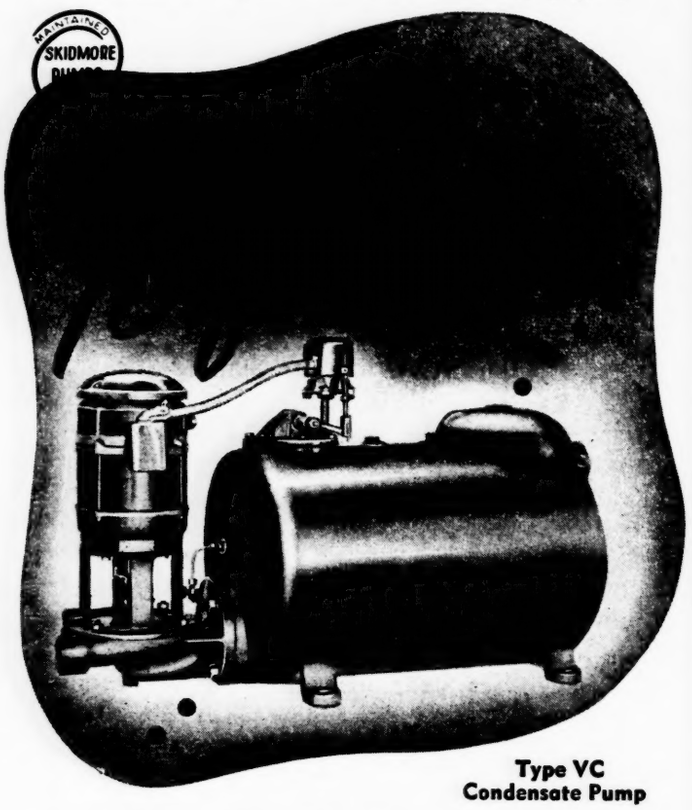
City.....Postal Zone.....State.....

Firm*.....Position*..... H&V-10/50

*This information would be appreciated for our private records.

Why Skidmore

- **Advanced Engineering**—Known as a leader in its field for many years, Skidmore Pumps are still first with the latest in pump engineering features.
- **Sturdy Construction** has made Skidmore Pumps a wise investment in the future, because you can depend on them to give years of dependable service.
- **Design Simplicity** means greater performance with lower maintenance. Entire motor and pump unit can be removed for inspection or repair without disturbing piping.



Type VC
Condensate Pump

• New Bulletin No. 10-B

Send for your copy of the Skidmore VC Condensation Pump bulletin which gives complete information and engineering data on this pump—or information on any other type Skidmore Pumps will be sent on request, without obligation.

Skidmore Pumps

ST. JOSEPH, MICHIGAN
Manufacturers of Quality Pumps
for Over a Quarter of a Century

prize for his discoveries concerning X-rays and was assigned to the "Manhattan Project," working on the design of the atomic bomb. Dr. Compton will be guest speaker at the noon luncheon on December 5th. The subject of his talk is "Contrast of East and West."

Tyler S. Rogers will report on "Insulation as a Function of Heating and Air Conditioning." Mr. Rogers, assistant to the executive vice president of Owens-Corning Fiberglas Corporation, is past president of The Producers Council, Inc., which is the national association for manufacturers of building materials and equipment, and is the author of the book "Plan Your House To Suit Yourself."

Of special interest is a scheduled report on "Building Forecasts for 1951" by John T. Anderson, Armco Steel Corp.

R. C. Hienton, general sales manager of The Cleveland Electric Illuminating Co., and Robert V. Main, manager of the Electric Home Bureau, will report on the Illuminating Company's heating sales promotion program.

• **SECOND DAY.**—A panel discussion of dealer problems will open the session on Wednesday morning, December 6th. Plans for the 1951 Indoor Comfort Conferences will be outlined by Guy A. Voorhees, technical secretary of the association.

The noon luncheon on Wednesday will be in honor of Dean Lorin G. Miller, Michigan State College. Activity reports of the various Association Committees and reports of the research work in cooperation with the University of Illinois will be presented at the Wednesday afternoon session.

$1000 AWARD COMPETITION

announced by publisher of Heating and Ventilating to give annual award for advertising managers.

The Industrial Press, publisher of HEATING AND VENTILATING, anounces the establishment of the Annual Industrial Press Award, consisting of $1,000 to be given to the industrial advertising man who, in the opinion of the judges, has made the best documented presentation to his management proving the need for a specific advertising program and showing how that program fills the need. The award is given in cooperation with the National Industrial Advertisers Association.

The competition is open to everyone actively engaged in advertising or marketing to the technical, trade or industrial fields. Membership in the N.I.A.A. is not a requirement for eligibility.

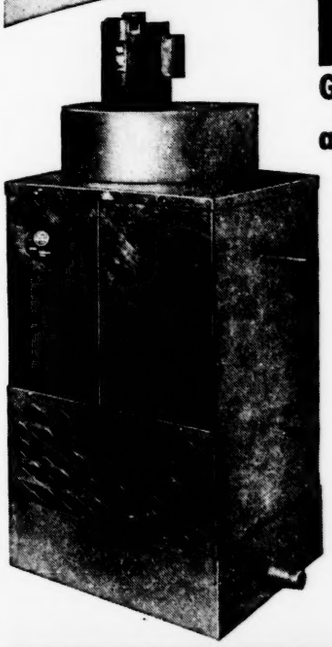
The purpose of The Industrial Press Award is threefold:

First, its object is to encourage advertising managers to undertake more thorough groundwork and to prepare more effective presentations to management establishing the need for a specific advertising program. If such a need is definitely established, an advertising program designed to fill that need would have manifest value and appeal.

Second, through the publicity given the Award and

SELL with ACME PRODUCTS MORE

ACME PRODUCTS



Get top quality and the biggest market

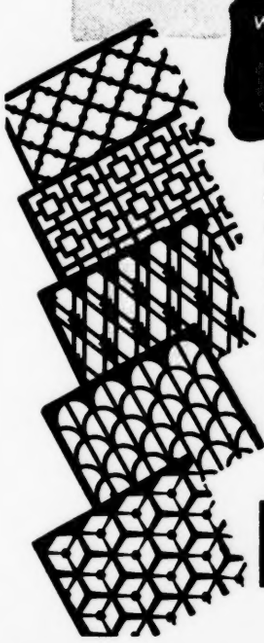
The new Acme Induced Cooling Towers open up new sales opportunities for you. Designed for use with self-contained refrigeration and air conditioning equipment, these packaged units are available up to and including 5 ton capacities. Can be used for outdoor or indoor application. All unnecessary parts have been eliminated to reduce weight to the minimum. Acme equipment has been serving the refrigeration industry since 1919 and the Acme name is your guarantee of scientific engineering backed by practical experience and acceptance in use. Address Dept. HV.

ACME INDUSTRIES INC.
JACKSON, MICHIGAN, U.S.A.

NEW JOBS OR REMODELING

GET MORE BEAUTY IN YOUR COMMERCIAL, RESIDENTIAL AND PUBLIC BUILDINGS *with*
H & K GRILLES

WIDELY USED FOR
- Air Conditioning
- Heating & Ventilating
- Radiator Enclosures, Etc.



H & K Grilles—a recognized standard in the field for over 65 years—are the solution to any enduring architectural plan. Here are patterns both modern and classical to suit every taste and purpose. Besides beauty, there are functional considerations, too... matters of safety, ventilation, screens for hidden controls and equipment. When you specify "Grilles by H & K", you're sure of outstanding workmanship in a wide choice of designs and materials.

Write for literature

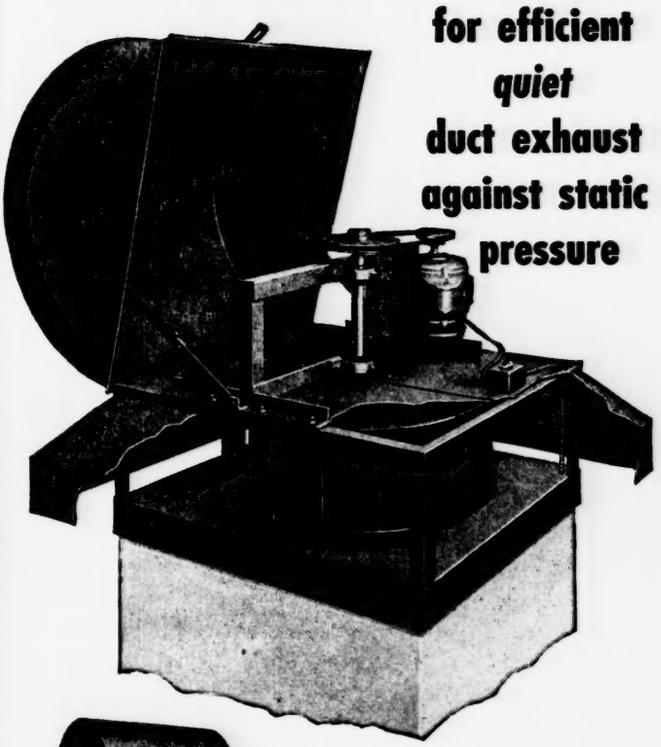
The Harrington & King Perforating Co.
5664 Fillmore Street
Chicago 44, Ill.

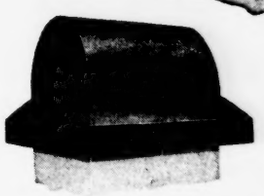
The New Swartwout

Airlift

Roof Ventilator

with certified fan ratings for efficient *quiet* duct exhaust against static pressure





This new duct exhaust ventilator brings you the quiet ventilation you've wanted for offices, schools, recreation halls and other applications. Airlift is powered by an unusually efficient centrifugal type fan unit having a streamlined inlet. Other structural features maintain Airlift's air moving effectiveness while operating at *very low noise levels.*

Airlift is made in a range of sizes and capacities covering all usual requirements. Powered by high quality motor; ball bearing fan shaft mounting; easily accessible for servicing.

For complete description of this new Airlift, write for Bulletin 341H.

18511 Euclid Avenue, Cleveland 12, Ohio.

The Swartwout Co.

Industrial Ventilation Specialists Since 1904

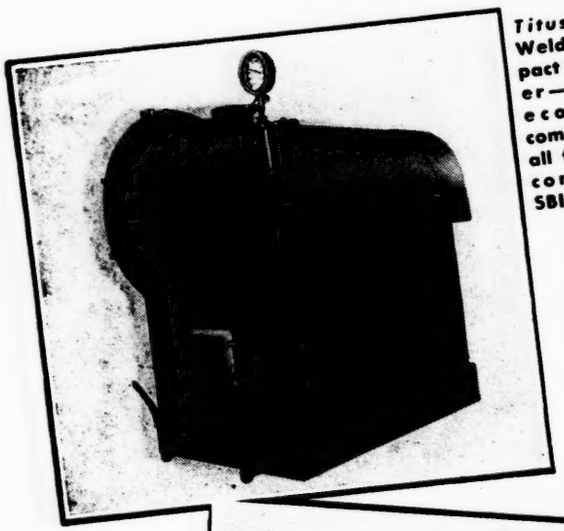
Equipment for *Controlled* Air Circulation

TITUSVILLE Builds

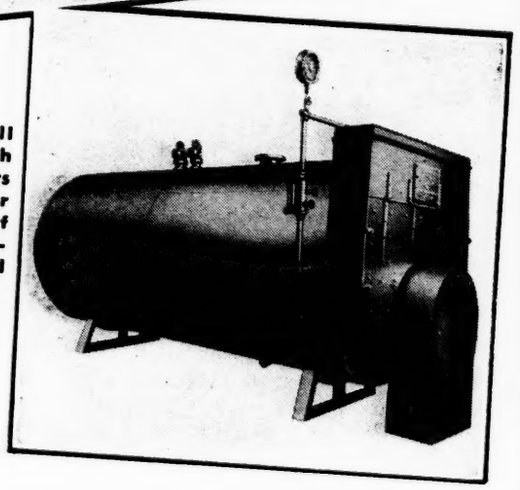
A COMPLETE LINE
OF DEPENDABLE, ALL WELDED

HEATING BOILERS

IN SIZES AND RATINGS TO FIT EVERY REQUIREMENT



Titusville All Welded, Compact Steel Boiler—provides economical combustion for all fuels. ASME constructed, SBI rated.



Titusville All Welded Scotch Heating Boilers—available for all types of fuel. ASME constructed, SBI rated.

Since 1860 TITUSVILLE BOILERS have been the accepted standard of quality, dependability and economy. Get the TITUSVILLE facts before purchasing any new or replacement boiler.

Write today for new technical, informative bulletins

The TITUSVILLE IRON WORKS Co.
DIVISION OF STRUTHERS WELLS CORPORATION
TITUSVILLE, PENNSYLVANIA

the announcement of the Award-winning entry, many top company officers will be impressed with the job that advertising can do.

Third, summaries of the presentations distributed to the salesmen would show them what the company advertising was setting out to accomplish, and where and how it would be directly beneficial to them.

Complete entry regulations and suggestions for those wishing to enter the competition may be obtained by writing The Industrial Press, 148 Lafayette Street, New York 13, N. Y.

The Industrial Press Award is not concerned with the preparation or execution of the advertising campaign, or with the results achieved, but rather with the groundwork which was done to establish the market pattern for a product, and the documentation used to show how a specific advertising program fits the requirements of this market pattern. The presentation must, of course, give an outline of advertising objectives, but details of the campaign itself and subsequent results of the advertising should not be covered in the entry.

The Awards Administrative Committee of the National Industrial Advertisers Association will have complete authority with regard to annual judging procedure, and will select a panel of judges representing men from industrial top managements who are not directly connected with advertising.

LARGEST ATOMIC FURNACE

for research begins operating at Brookhaven National Laboratory.

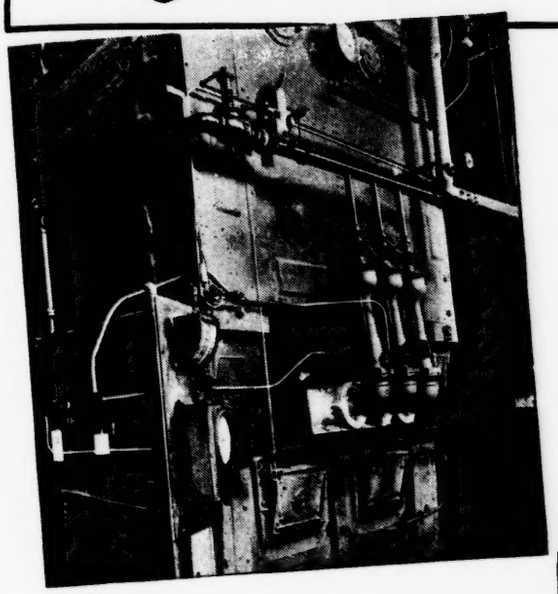
The nation's largest and most powerful research reactor began operating August 22 at Upton, N. Y., according to Dr. Leland J. Haworth, director of Brookhaven National Laboratory, nuclear research center supported by the Atomic Energy Commission.

At 2:30 a.m. Dr. Lyle B. Borst, in charge of the reactor (atomic pile), signalled for removal of control rods, thereby starting the initial chain reaction. Loading of uranium to bring the reactor to its full designed power will continue for several months. Meanwhile, scientific experiments will begin, with the reactor operating 24 hours a day.

The "atomic furnace" is an air-cooled, graphite-uranium pile which, at its designed power level, will develop heat at a rate of 30,000 kilowatts, several times that of the AEC reactor at Oak Ridge National Laboratory. Engineering design and construction were carried out by The H. K. Ferguson Company of Cleveland and New York, under the supervision of Wells N. Thompson, vice president, acting on scientific specifications established by the Brookhaven reactor staff. The contract under which Brookhaven operates is supervised by Wilbur E. Kelley, manager of the New York Operations Office of the AEC, and E. L. Van Horn, manager of AEC's Brookhaven area.

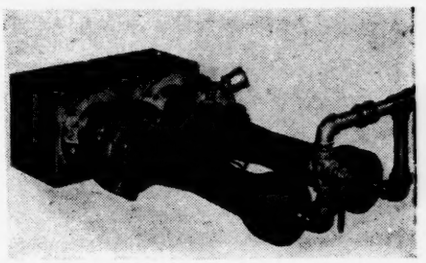
• **COOLING SYSTEM.** — Passageways in the graphite cube of the reactor permit circulation of cooling air to carry away the tremendous heat generated by fission, thus preventing the reactor from overheating.

Bryant Industrial TORCH BURNER ASSEMBLIES



Three Bryant Block Torches applied to 150 HP water tube boiler, using 1½ PSIG-Gas.

FOR GAS-FIRING BOILERS, DRYERS, BAKE OVENS, AIR HEATERS



Especially designed for low and medium-pressure gas without using blowers and utilizing air supplied by draft. Made in nine sizes, inputs up to 7,000,000 Btu per hour. Simple, easy to install and control. For complete information, write for Data Sheet 1A-5b. Bryant Industrial Division, Affiliated Gas Equipment, Inc., 17825 St. Clair Avenue, Cleveland 10, Ohio.

GAS Bryant Industrial Division COMBUSTION FOR INDUSTRY AFFILIATED GAS EQUIPMENT, INC.

Over 2,000 Plants Find by Experience that

DACAROL
FUEL OIL TREATMENT SAVES MONEY!

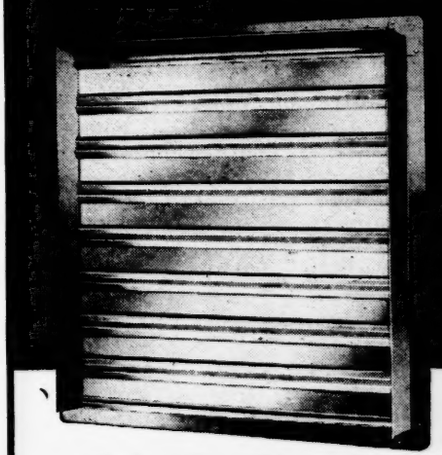
DACAROL is used successfully in more than 2,000 large industrial plants and is now available for homes and small plants. DACAROL is an oil-soluble penetrating and dispersing additive which penetrates the entire body of oil within a few hours, disperses sludge, produces more efficient atomization, and gives greater Btu release. Result: You save fuel and decrease operating costs! Merely add to oil in tank—no agitation required. Non-explosive, harmless to hands, metals, piping, gaskets, fittings.

WRITE FOR complete data and engineering service.

Also Water Treating Specialists.
Formulated blends of chemicals for specific problems.
Registered Professional Engineers

Dacar Chemical Products Co.
McCartney Street **Pittsburgh 20, Pa.**

they all want AIR-FLO



MODEL A-555

THE AUTOMATIC SHUTTER WITH ALL THE FEATURES

Front View—Closed

IT TAKES THE LOAD OFF THE FAN!

Aluminum louvers open fully, permitting capacity fan operation. New heavy reinforcement strip adds strength and long life to the louvers, assures quiet operation and perfect counterbalance, prevents rattling. Deep shroud protects shutter from high winds. Tie-rod, brackets and bearings inside frame, not exposed to weather. Special finish resists corrosion. Many other features.

Write for New Air-Flo Catalog 43-F
Illustrations and details of the complete Air-Flo line.

AIR CONDITIONING PRODUCTS CO.
2340 W. LAFAYETTE BLVD. · DETROIT 16, MICH.

for • SPRAYING
• WASHING
• RINSING
• COOLING
• AIR CONDITIONING

YARWAY

SPRAY NOZZLES

FAN-TYPE FOR FLAT SLICING SPRAY

Get the most out of your Spraying Equipment with minimum power . . . with efficient spraying.

Use Yarway Nozzles. No internal vanes or other restrictions to clog or hinder flow. Two types—Yarway Involute-type producing a fine hollow spray with minimum energy loss, and Yarway Fan-type producing a flat fan-shaped spray with time-saving slicing action for cleaning.

Wide range of standard sizes and capacities. Cast or machined from solid bar stock.

Thousands in use. - - Write for Bulletin N-616.

YARNALL-WARING COMPANY
104 Mermaid Avenue Philadelphia 18, Pa.

Under and around the reactor are labyrinthine passages, or ducts, through which cooling air enters and leaves. Air speed varies from some hundreds of miles per hour in the narrow graphite passageways to 40 miles per hour in the ducts.

In the nearby fan house, specially designed centrifugal compressors rotate at a speed of 3,600 revolutions per minute to draw 300,000 cu ft of air every minute through the pile cooling system. Each 12-foot blower is driven by a 1,500 horsepower motor. After the cooling air has been circulated through the reactor, it passes through an air duct to the fan house, where it is exhausted up the 320-foot stack.

One natural constituent of the air used to cool a reactor is the inert gas argon, which comprises about one percent of the earth's normal atmosphere. In passing through the reactor a part of this gas becomes radioactive. However, this radioactivity immediately begins decaying. In addition, it is diluted many thousands of times after it leaves the stack, by a process of mixing with the natural atmosphere. Decay and dilution together render the radioactivity harmless.

If the cooling air carried particles of dust, some of this material, too, might become radioactive. To avoid this, the air is filtered both before and after it passes through the reactor.

• **METEOROLOGY.**—If weather conditions threaten to cause a poorly diluted flow of reactor gases over one spot long enough to raise the radiation exposure above the permissible weekly limit at ground level, the reactor will be shut down. To provide detailed technical information, the Meteorology Group attached to the reactor operates a weather station and two towers, 420 and 160 feet high. Meteorologists measure temperature and wind differences between ground and the tops of the towers. Thus they can determine how conditions affecting rates of dilution vary.

NEWS BRIEFS

• Leonard C. Bastian, technical director of the Air Conditioning and Refrigerating Machinery Association during the past twelve years, has joined the engineering staff of Philco Corporation. J. R. Schreiner, formerly associated with the research laboratory of Pennsylvania State College, has joined the ACRMA staff at its Washington headquarters as staff engineer.

• Promotion of Roy W. Maze to the position of merchandising sales manager of The Marley Co., Inc., has been announced. In 1946, Mr. Maze joined The Marley Company as public relations director, the position he held until his recent promotion.

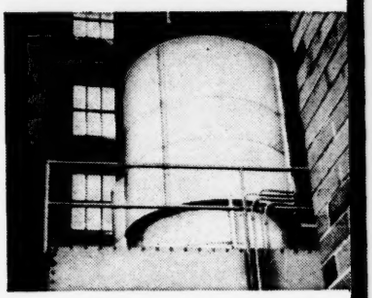
• United States Air Conditioning Corp., Minneapolis, has announced the appointment of Marvin M. Yarosh as head of usAIRco's test laboratory. For the last two years, while doing graduate work at the University of Minnesota, Mr. Yarosh was instructing and doing research.

Industrial Sound Control

QUIETS

COOLING TOWERS

in Air Conditioning Installations



Ends

- **ANNOYANCE**
- **COMPLAINTS**
- **THREATS OF LEGAL ACTION**

due to disturbing noise

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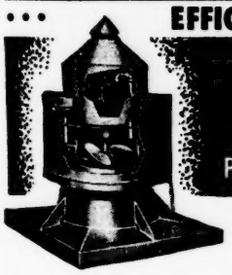

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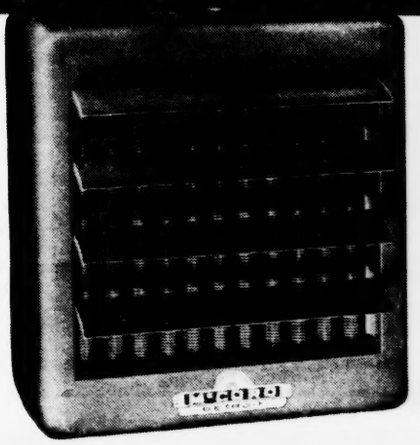

Modern design—top performance—thousands in use—dependable heat. All copper coils to resist corrosion — individual tube expansion — most rugged construction — quiet operation — easy installation—standard motors with standard bases. Sizes to 272,000 B.T.U. capacity.

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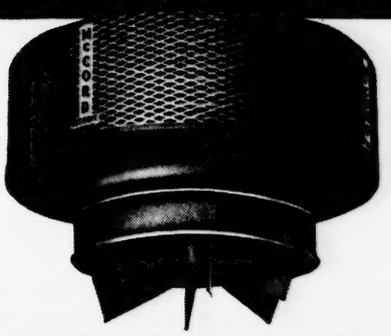

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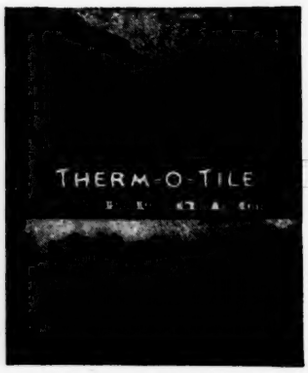

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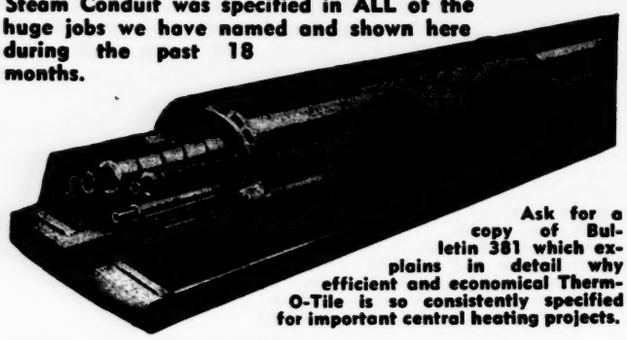

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ABSTRACTS AND REVIEWS

Publications abstracted in this department should be ordered direct from publisher.

PROCESS HEAT TRANSFER

The academic subject of heat transfer is related to industrial practice in Process Heat Transfer, by Donald Q. Kern, adjunct professor of chemical engineering, Polytechnic Institute of Brooklyn. Here are combined both theoretical background and modern practice. Solutions are given for the numerous representative problems arising in the chemical and mechanical process industries.

A number of empirical calculation methods are given which the author claims has not previously appeared in engineering literature. An effort has been made for simplification.

To hold down the size of the book, the author has omitted material which, while customarily included in many books, is not ordinarily used in the solution of common engineering problems.

The subject is thoroughly covered through the explanation of basic theory and the solution of typical problems.

The titles of some of the 21 chapters are: shell and tube exchangers; flow arrangement for increased heat recovery; gases; streamline flow and free convection; calculations for process conditions; condensation; evaporation; vaporizers, evaporators and reboilers; extended surfaces; cooling towers; batch and unsteady state processes; furnace calculations; control of temperature and related process variables.

Process Heat Transfer, by D. Q. Kern. Cloth bound,

6 x 9 inches, 871 pages. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York 18, N. Y. Price, $8.

HEAT AND TEMPERATURE MEASUREMENT

Descriptions of methods of heat and temperature measurement, and the theoretical principles necessary for their appreciation, are presented in Heat and Temperature Measurement, by Robert L. Weber, associate professor of physics, The Pennsylvania State College.

The book is divided into two parts. Part 1 covers the physical principles which serve as a basis for thermal measurements. There is a concise chapter summary at the end of each chapter to review the fundamental ideas. Emphasis is on experimental methods rather than on thermodynamic theory. Part 2 outlines the procedures for 29 laboratory experiments. Brief discussions are given of the theory back of the laboratory approach.

The material is arranged so that it is suitable for a one-semester college course. The following chapters cover the first part of the book:

Temperature scales; expansion thermometers; heat transfer; thermoelectricity; thermoelectric temperature measurements; radiant energy; optical and radiation pyrometry; resistance thermometry; temperature recording and control; calorimetry; phase diagrams, thermal analysis; thermodynamics; extreme temperatures; special methods of temperature measurement; international temperature scale.

Heat and Temperature Measurement, by R. L. Weber. Cloth bound, 5½ x 8½ inches, 422 pages. Published by Prentice-Hall, Inc., 70 5th Ave., New York 11, N. Y. Price, $6.65.

• • •

CORROSION OF HOT WATER STORAGE HEATERS—The AGA research project, "Galvanic Corrosion of Dissimilar Metals as Applied to Gas Hot Storage Heaters," has been completed by Dr. Robert C. Weast of Case Institute of Technology, and the final data, summary and conclusions are contained in Part III of a committee report.

This project, DGR-3-WH, was sponsored by the Committee on Domestic Gas Research, R. J. Rutherford, chairman. Technical supervision was by the Technical Advisory Group for Gas Water Heating Research, John W. Farren, chairman, and A. F. Craver represented the Group as Project Technical Advisor. American Gas Association, 420 Lexington Ave., New York 17, N. Y. Price, 50 cents.

RESEARCH WORK ON COAL—Summarizing many of the coal publications issued by the U. S. Bureau of Mines, the 14th resume of the research conducted on coal is now available. This resume covers changes and advance in coal mining technique, drilling investigations, etc. The report has many illustrations and references to coal publications issued by the Bureau of Mines. For a copy of Information Circular 7565 on Annual Report of Research and Technologic Work on Coal, Fiscal Year, 1949, write to Bureau of Mines, Publications Distribution Section, 4800 Forbes St., Pittsburgh 13, Pa.

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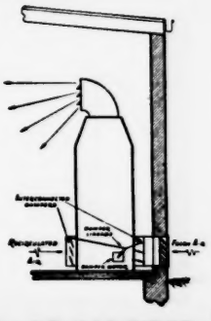

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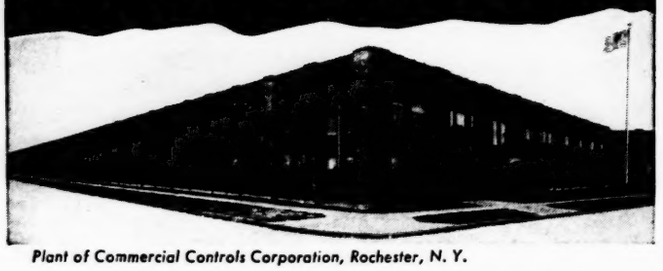

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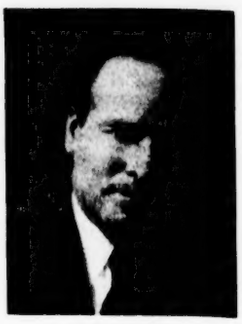

A. J. Nesbitt

Albert J. Nesbitt (*Heating and Ventilating Schools with Unit Ventilators*, page 79) president of John J. Nesbitt, Inc., has been associated with that company since 1917. He attended the public schools of Atlantic City, N. J., and Drexel Institute of Technology, Philadelphia, Pa. He is a professional engineer registered in Pennsylvania, New Jersey and New York.

During World War II, Mr. Nesbitt served as an Industry Member of the Regional National War Labor Board, and on the Industry Advisory Committee of the War Production Board. In a civilian capacity he served as assistant chairman of the Industry Integration Committee of the Ordnance Department under Brig. Gen. R. E. Hardy, and at the conclusion of the war was awarded a citation by the War Department for his services.

A past president of the Industrial Unit Heater Association, he holds membership in the ASHVE and the National Society of Professional Engineers. He is the author of many articles on schoolhouse heating and ventilating.

Mr. Nesbitt is active in national committees of the YMCA and is a member of the board of directors of the Community Chests and Councils of America.

COMING EVENTS

Where listed, names or titles of individuals are those from whom further information is available.

ASME CONFERENCE—Fuels Division conference of the American Society of Mechanical Engineers, at Hotel Statler, Cleveland, Ohio. C. E. Davies, secretary of the Society, 29 W. 39th St., New York, N. Y. .....**OCTOBER 23-25, 1950.**

METALS MEETING—Annual meeting of the American Society for Metals, in Chicago, Ill. For hotel and other details, write W. H. Eisenman, national secretary, 7301 Euclid Ave., Cleveland 3, Ohio. .....**OCTOBER 23-27, 1950.**

WELDING MEETING—Annual meeting of the American Welding Society, in Chicago, Ill. For hotel and other details, write J. G. Magrath, national secretary, 33 W. 39th St., New York 18, N. Y. .....**OCTOBER 23-27, 1950.**

PETROLEUM MEETING—30th annual meeting of the American Petroleum Institute, at the Hotel Biltmore and Hotel Ambassador, Los Angeles, Calif. Secretary of the Institute, 50 W. 50th St., New York, N. Y. ....**NOVEMBER 13-16, 1950.**

REFRIGERATION WEST COAST EXHIBIT—West Coast Educational Exhibit and Conference, of the Refrigeration Equipment Manufacturers Association, at Municipal Auditorium in Long Beach, Calif., to be held simultaneously with 5th annual convention of the Refrigeration & Air Conditioning Contractors Association. Edna Berggren, executive secretary, 228 No. La Salle St., Chicago 1, Ill. .....**NOVEMBER 17-19, 1950.**

ASME MEETING—Annual meeting of the American Society of Mechanical Engineers, at the Hotel Statler, New York, N. Y. C. E. Davies, secretary of the Society, 29 W. 39th St., New York, N. Y. .....**NOVEMBER 26—DECEMBER 1, 1950.**

POWER SHOW — 19th National Exposition of Power and Mechanical Engineering, sponsored by the American Society of Mechanical Engineers, in conjunction with the society's annual meeting, at Grand Central Palace, New York, N. Y. Charles F. Roth, president, International Exposition Company, Grand Central Palace, New York.....**NOVEMBER 27—DECEMBER 2, 1950.**

ASRE MEETING—Annual convention of the American Society of Refrigerating Engineers, at Hotel Commodore, New York. M. C. Turpin, secretary, 40 W. 40th St., New York 18, N. Y. .....**DECEMBER 3-6, 1950.**

AUTOMOTIVE ENGINEERS MEETING—Annual meeting and engineering display, of the Society of Automotive Engineers, at the Hotel Book-Cadillac, Detroit, Mich. John A. C. Warner, secretary of the Society, 29 West 39th Street, New York, N. Y. .....**JANUARY 8-12, 1951.**

PLANT MAINTENANCE SHOW AND CONFERENCE—2nd plant maintenance show and conference, at the Auditorium, Cleveland, Ohio. Clapp & Poliak, Inc., 341 Madison Ave., New York 17, N. Y. .....**JANUARY 15-18, 1951.**

ASHVE MEETING—57th annual meeting of the American Society of Heating and Ventilating Engineers, at the Bellevue-Stratford, Philadelphia, Pa. A. V. Hutchinson, secretary of the Society, 51 Madison Ave., New York 10, N. Y. .....**JANUARY 22-25, 1951.**

ASHVE EXPOSITION—10th International Heating and Ventilating Exposition under the auspices of the American Society of Heating and Ventilating Engineers, at the Commercial Museum, Philadelphia, Pa. Charles F. Roth, manager of the International Exposition Company, Grand Central Palace, New York 17, N. Y. .....**JANUARY 22-26, 1951.**

ASME MEETING—Spring meeting of the American Society of Mechanical Engineers, at the Hotel Atlanta-Biltmore, Atlanta, Ga. C. E. Davies, secretary of the Society, 29 W. 39th St., New York, N. Y. .....**APRIL 2-5, 1951.**

FOUNDRYMEN'S CONVENTION—55th annual convention of the American Foundrymen's Society, in Buffalo, New York. William W. Maloney, secretary-treasurer of the Society, 616 S. Michigan Ave., Chicago 5, Ill. .....**APRIL 23-26, 1951.**

REFRIGERATION MEETING—Annual meeting of the Refrigeration Equipment Manufacturers Association, at The Homestead, Hot Springs, Va. George E. Mills, public relations director, REMA, 1346 Connecticut Ave., N.W., Washington, D. C. .....**APRIL 30—MAY 2, 1951.**

ARCHITECTS CONVENTION—1951 convention and national seminar meetings of The American Institute of Architects, at the Edgewater Beach Hotel, Chicago, Ill. Paul Gerhardt, Jr., chairman, 121 No. LaSalle St., Chicago, Ill. **MAY 8-11, 1951.**

NDHA MEETING—42nd annual meeting of the National District Heating Association, at the Statler Hotel, Detroit, Mich. John F. Collins, Jr., secretary-treasurer of the Association, 827 N. Euclid Ave., Pittsburgh 6, Pa.....**JUNE 5-8, 1951.**

INSTRUMENT CONFERENCE AND EXHIBIT—1951 Instrument conference and national exhibit of the Instrument Society of America, at the Coliseum, Houston, Tex. John McCaffery, assistant executive secretary of the Society, 921 Ridge Ave., Pittsburgh 12, Pa. .....**SEPTEMBER 10-14, 1951.**

REFRIGERATION SHOW—7th All-Industry Refrigeration and Air Conditioning Exposition, at Navy Pier, Chicago, Ill. Show director, REMA, 1346 Connecticut Ave., N.W., Washington, D. C. .....**NOVEMBER 5-9, 1951.**

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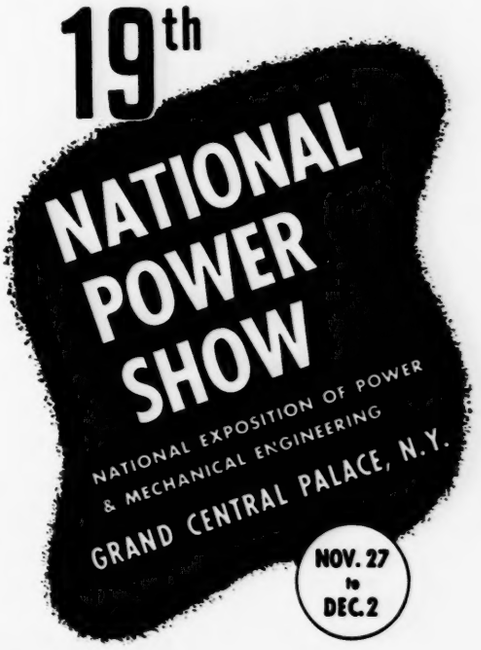

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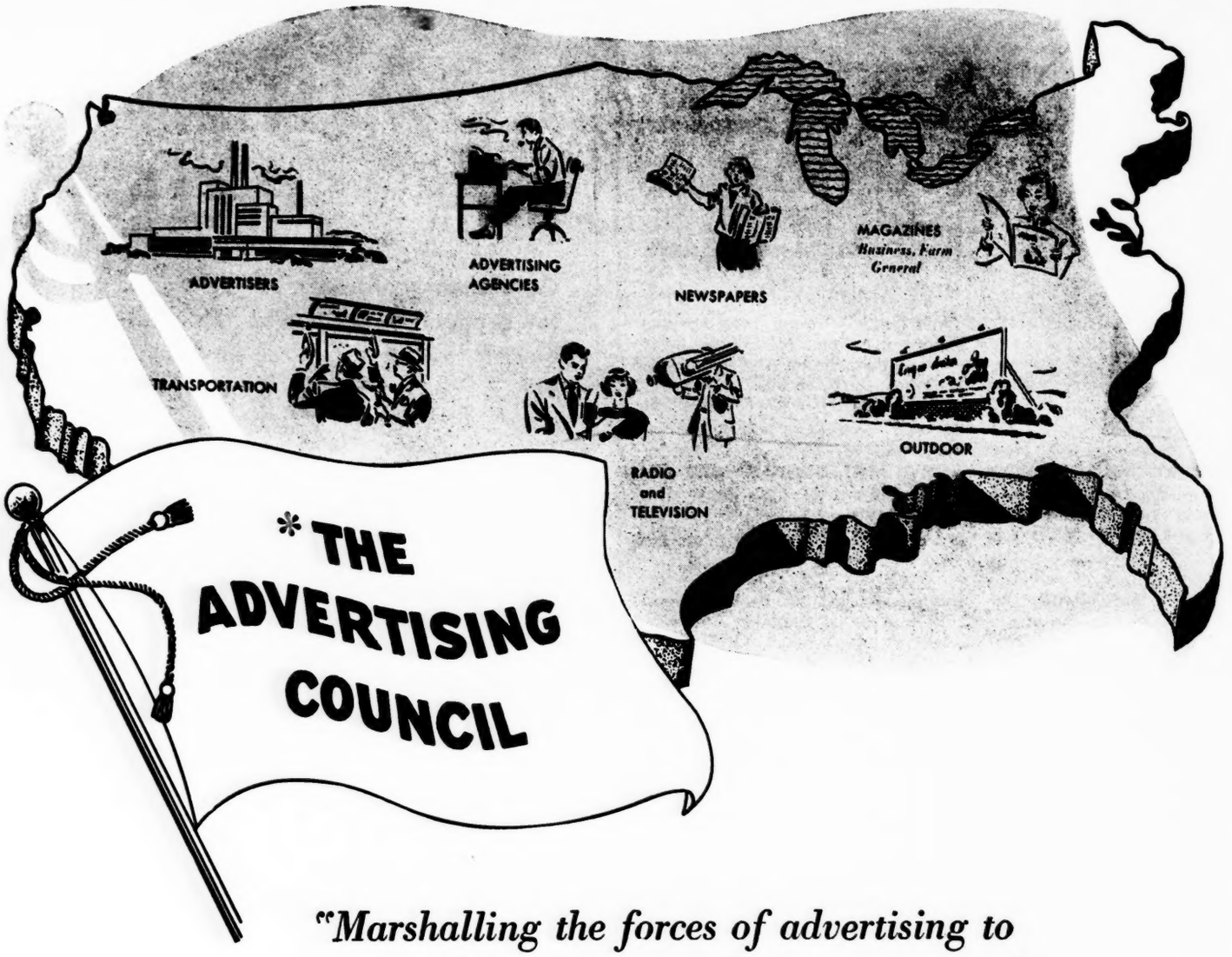

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GLOSSARY OF TERMS USED IN PLUMBING

Backflow. Water flow into water-distributing pipes from any source other than that for which the system is designed.

Branch, Fixture. The branch from the water-distributing pipe in the building to the fixture supply pipe.

Branch, Horizontal. The drain line from the fixture drain to the waste stack.

Closet, Frost-proof. One that has no water in the bowl and with the trap and water supply valve below the frost line.

Conductor. Same as Leader.

Connection, Backflow. Any connection or arrangement which permits backflow.

Connection, Cross. Any arrangement or connection by which the water supply can be contaminated.

Downspout. Same as Leader.

Drain. A pipe carrying waste water in a building drainage system.

Drain, Area. One installed to collect rain or surface water from an open space.

Drain, Building. The section of the horizontal drain pipe connecting the drainage pipes inside the building with the building sewer 5 feet outside the inner face of the building wall.

Drain, Building Storm. A drain for carrying rain, surface water, condensate, or similar discharge to the building drain or sewer.

Drain, Fixture. The drain from the trap of a fixture to the point where it connects with another drain pipe.

Drain, House. See Drain, Building.

Drain, Roof. One on the roof to collect water on the roof surface and discharge it to leader.

Drain, Subsoil. One for collecting sub-surface or seepage water and carrying it to a point of disposal.

End, Dead. A branch from a building sewer, vent pipe or soil pipe which is capped or plugged 2 feet or more from the sewer or vent pipe along its developed length.

Fixture, Combination. A combined sink and laundry tray(s).

Fixtures, Plumbing. Devices which receive or make available water, or receive liquid or water-borne wastes, and discharge the wastes into a drainage system.

Flooded. A situation when the liquid in a vessel or fixture rises to the flood level.

Gap, Air. The distance, vertically, from a faucet opening or pipe opening through the air to the flood-level rim of the fixture.

Grade. In plumbing, the slope of a line compared to horizontal, usually expressed in inches fall per foot of pipe.

Interceptor, Grease. A trap designed to collect and retain grease and fats from kitchen waste lines, and installed in the line between the kitchen and building sewer.

Interceptor, Sand. A trap designed to retain sand.

Interval, Branch. A vertical length of waste pipe usually one-story high and to which the horizontal branches from that floor are connected.

Leader. A water conductor from the roof to the storm drain.

Length, Developed. The length of a section of pipe along the center line through the pipe and fittings.

Level, Flood. That level at which water begins to overflow the rim of the fixture.

Main, Water. Street main.

Offset. An arrangement of fittings or bends changing the line of piping to another line but parallel to the original line.

Offset, Double. Two offsets in succession in the same line.

Offset, Return. A double offset is an arrangement which offsets the line, then returns it to the original line.

Opening, Effective. The smallest cross-sectional area in the water-supply discharge in terms of the diameter of the circle of a round opening or, if the opening is not round, the diameter of a circle having that cross-sectional area.

Pipe, Drain. Same as Drain.

Pipe, Fixture Supply. The pipe connecting the fixture with the fixture branch at the floor or wall line.

Pipe, Indirect Waste. One which drains through a trapped fixture or receptacle to the drainage system.

Pipe, Soil. One which carries the discharge from water-closets or fixtures.

Pipe, Special Waste. See Pipe, Indirect Waste.

Pipe, Vent. See System, Vent.

Pipe, Waste. A line carrying the discharge from any fixture except water-closets and the like, and which conveys the waste to the building drain or waste stack. If it is not directly connected to the building drain or soil pipe it is an Indirect Waste Pipe.

Pipe, Water Distributing. One carrying water from the water service pipe to the fixtures.